W. B. Cooley
May 4, 1905

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MECHANICS

OF

AIR MACHINERY

BY

DR. JULIUS WEISBACH

AND

PROFESSOR GUSTAV HERRMANN

AUTHORIZED TRANSLATION,

WITH AN APPENDIX ON AMERICAN PRACTICE

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TRANSLATOR'S PREFACE.

THIS volume contains a translation of the portion of Professor Herrmann's revised edition of Weisbach's work on Engineering Mechanics pertaining to the Moving of Air.

It is the final portion of Weisbach's work to be published in English, and is supplemented by an appendix showing some features of recent American practice in Air Machinery. In this appendix there has been included a very comprehensive description of the most up-to-date machinery for moving air. This portion of the book will be found useful for the designing of machinery that is to meet modern conditions, as the demands that are made by users of air machinery are indicated, and some of the devices, developed to meet these demands, are shown. Care has been taken to eliminate as much as possible descriptions of engines or other methods of driving the air machinery, but wherever it was found necessary these have been mentioned. The changes made necessary by the introduction of high-speed steam-engines and gas-engines are described, as are also the various kinds of fan-blowers that have come into such common use in the last decade.

The notes from which the translation has been prepared were made by Professor J. F. Klein of Lehigh University.

References to other portions of Weisbach's Mechanics are to the English editions.

INTRODUCTION.

THE earliest form of machine for moving air was unquestionably a fan moved back and forth by hand, but this was very quickly succeeded by the bellows, and that form of air machine has continued in use to the present day. While the bellows has been in use all these centuries, it is not because of its efficiency, but rather because of its extreme simplicity and cheapness. The question of efficiency is seldom considered in connection with a bellows, as the clearances are necessarily so great that not even a fair efficiency, compared with later forms of blowers, can be obtained. As the bellows is so familiar to everybody, no description of it will be given in these pages, and the descriptions of many of the old forms of air machines and their driving mechanisms given by Weisbach will also be omitted.

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MECHANICS OF AIR MACHINERY.

THE MOVING OF AIR.

§ 1. General Remarks on the Moving of Air.—Air can be moved from a point A to another point B, Figs. 1 and 2, either by increasing its expansive force at A or by diminishing the expansive force at B. Let p, γ, and t represent the pressure,

FIG. 1.

FIG. 2.

density, and temperature of the air at A; and p_1, γ_1, and t_1 the same quantities for the air at B; then, according to the combined laws of *Gay-Lussac* and *Mariotte* (Vol. I, sec. VI, chap. 4),

$$\frac{p_1}{p}=\frac{1+0.00367t_1}{1+0.00367t}\frac{\gamma_1}{\gamma}\ (\text{C.}\ ^\circ)\quad \left(\frac{p_1}{p}=\frac{1+0.00203(t_1-32)\gamma_1}{1+0.00203(t-32)\gamma}\text{Fah.}\ ^\circ\right).$$

The difference $p-p_1$ needed to move the air from A to B can therefore be obtained either by varying the *temperature* or the *density* γ.

Accordingly there are two methods of moving air, namely,

(1) By heating or cooling one side;

(2) By making the air at one side dense or rare (i.e., by compressing or expanding it).

Among the means employed in the first method are the fireplaces of ordinary heating apparatus and the air-furnaces in mines with their adjuncts, as chimneys, flues, air-shafts, etc. Among the means employed in the second method are the ventilating-machines and blowers used in mines and in metallurgical establishments. The ventilating-machines used in mines are usually *air-exhausters,* i.e., they cause the air to move from A to B by rarefying the air in B, while the blowers employed by metallurgists are *air-compressors,* which drive the air from A to B by making it more dense at A. The object of ordinary ventilating-machines is to create a current of air to replace that vitiated by respiration or combustion, while the object of blowers is to force the air with increased pressure and great velocity into the smelting- or combustion-chamber of a furnace. In other respects there is no essential difference between blowers and exhausters, for generally one machine can be transformed into the other by changing its position or some of its parts, for example the valves. Thus the ventilating-machine or suction-pump C, Fig. 2, differs from the blower or force-pump C, Fig. 1, simply in having a different position relatively to the receivers A and B.

The compression and expansion of air in ventilating-machines and blowers can be effected either by means of a solid or fluid body, the latter being usually water. The solid body works either like a pump-piston, with a reciprocating or continuous rotary motion, enlarging or contracting a certain space, or by communicating a great velocity to the air by a rapid rotation, the inertia thus acquired rendering the air to be moved more dense or rare. These latter are known as *centrifugal blowers* or *pumps,* and they resemble centrifugal pumps, while those first mentioned correspond to either reciprocating- or piston-pumps. One more difference may be mentioned, namely, that the piston-like body is *packed* by a rigid or fluid material so as to make it air-tight, or the change of volume is effected without packing by means of a flexible material so as to prevent leakage, as in leather *bellows.*

Finally, the blowers which compress and move the air by

the aid of water are of various construction; they are principally represented by the *spiral* chain, *water-pressure* and *water-jet* blowers. Steam-jets have also recently been employed as blowers.

We shall now discuss, in the order above indicated, the most important of these arrangements and machines. It is plain that many of the pumps and water-raising machines discussed in the Mechanics of Pumping Machinery can be, directly or with slight modifications, used for moving air.

§ 2. The Moving of Air by Difference of Temperature.—The simplest case of moving air is seen in the ordinary *ventilation* of dwellings and the *natural ventilation* of mines. The air in a pipe, *ABCD,* Figs. 3 and 4, is set in motion whenever its temperature differs from the outside air, if the ends of the pipe are in communication with the outside air at different levels.

FIG. 3.

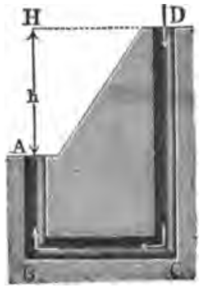

FIG. 4.

If h is the perpendicular distance AH between the levels of the orifices at A and D, t the temperature of the outer, and t_1 that of the inner air; then, according to Volume II (Theory of Chimneys), we have for the theoretical velocity of the air at D either

$$v=\sqrt{\frac{\delta(t_1-t)}{1+\delta t}2gh}$$

$$=\sqrt{\frac{0.00367(t_1-t)}{1+0.00367t}2gh}* \left(v=\sqrt{\frac{0.00203(t_1-t)}{1+0.00203(t-32)}2gh}\right),$$

* The formula is found as follows:

Let W = weight of air flowing per second through orifice of efflux;
F = area of orifice of efflux;
V = volume of air flowing per second through orifice of efflux;

or

$$v=\sqrt{\frac{\delta(t-t_1)}{1+\delta t}2gh}$$

$$=\sqrt{\frac{0.00367(t-t_1)}{1+0.00367t}2gh}\ \left(v=\sqrt{\frac{0.00203(t-t_1)}{1+0.00203(t-32)}2gh}\right).$$

The first formula being employed when the inner temperature t_1 is the greater, the air flowing out of the highest point D (Fig. 3), and the second formula when *this* temperature is less than the outer temperature, the air flowing out of the lower orifice A (Fig. 4). The velocity of the air in the conduit $ABCD$ therefore increases not only with the square root of the difference of level h of the orifices A and D, but also as the square root of the difference of temperature $(t-t_1)$ or (t_1-t). This velocity is reduced by the resistances to motion in the conduit, particularly by the friction at the sides. If l is the length of the whole conduit $ABCD$, $d=\frac{4F}{p}$ (see Vol. I), its hydraulic mean diam., $\zeta=0.024$, the coefficient of friction of the air, and if we represent the sum of the coefficients of all the other resistances in the pipe by ζ_1, then, assuming the cross-section of the pipe to be everywhere the same, and consequently that the orifices are equal, we have for the velocity of the air in the conduit

---

Let p = net driving pressure per unit of area of orifice of efflux $=h(\gamma-\gamma_1)$;

γ = specific weight of the outside air;

γ_1 = specific weight of the air at the orifice of efflux;

v = velocity per second of the air at the orifice of efflux;

then

$\frac{1}{2}\left(\frac{W}{g}\right)v^2$ = kinetic energy acquired by the air that passes out of the orifice;

$pV=p(Fv)$ = work imparted to air per second by driving pressure pF.

$$\frac{1}{2}\left(\frac{W}{g}\right)v^2=pV,\qquad \frac{v^2}{2g}\frac{W}{V}=p,$$

$$\frac{v^2}{2g}\gamma_1=h(\gamma-\gamma_1),\quad\text{or}\quad v=\sqrt{2gh\left(\frac{\gamma}{\gamma_1}-1\right)}.$$

Now since $\frac{\gamma}{\gamma_1}=\frac{1+\delta t_1}{1+\delta t}\frac{p}{p_1}$ and since p is nearly equal to p_1, the above equation follows.

$$v=\sqrt{\frac{\delta(t_1-t)2gh}{(1+\delta t)\left(1+\zeta\frac{l}{d}+\zeta_1\right)}}=\sqrt{\frac{0.00367(t_1-t)2gh}{(1+0.00367t)\left(1+0.024\frac{l}{d}+\zeta_1\right)}}$$

$$\left(v=\sqrt{\frac{0.00203(t_1-t)2gh}{1+0.00203(t-32)1+0.024\frac{l}{d}+\zeta_1}}\right),$$

or if we place $1+0.00367t=1$, we have approximately

$$v=0.0606\sqrt{\frac{(t_1-t)2gh}{1+0.024\frac{l}{d}+\zeta_1}}=0.268\sqrt{\frac{(t_1-t)h}{1+0.024\frac{l}{d}+\zeta_1}}\text{ meters}$$

$$\left(v=0.045\sqrt{\frac{(t_1-t)2gh}{1+0.024\frac{l}{d}+\zeta_1}}=0.362\sqrt{\frac{(t_1-t)h}{1+0.024\frac{l}{d}+\zeta_1}}\right).$$

If the pipe or conduit sides are very rough, ζ is greater than 0.024, and it may be well for safety to assume it equal to 0.05, as was done in Vol. II when calculating the velocity of air in chimneys.

If F denotes the cross-section of the pipe, the quantity of air per second flowing through it is

$$Q_1=Fv=0.0606F\sqrt{\frac{(t_1-t)2gh}{1+0.024\frac{l}{d}+\zeta_1}}$$

$$\left(Q_1=0.045F\sqrt{\frac{(t_1-t)2gh}{1+0.024\frac{l}{d}+\zeta_1}}\right),$$

and reducing this to the outer temperature t, we have

$$Q=\frac{1+\delta t}{1+\delta t_1}Q_1,$$

or approximately

$$\begin{aligned}Q&=[1-\delta(t_1-t)]Q_1\\&=F[1-\delta(t_1-t)]\sqrt{\frac{\delta(t_1-t)2gh}{(1+\delta t)\left(1+\zeta\frac{l}{d}+\zeta_1\right)}}\end{aligned}$$

$$=F[1-\delta(t_1-\tfrac{1}{2}t)]\sqrt{\frac{\delta(t_1-t)2gh}{1+\zeta\frac{l}{d}+\zeta_1}}$$

$$=0.0606F[1-0.00367(t_1-\tfrac{1}{2}t)]\sqrt{\frac{(t_1-t)2gh}{1+0.024\frac{l}{d}+\zeta_1}}$$

$$\left(Q=0.045F[1-0.00203(t_1-\tfrac{1}{2}t+16)]\sqrt{\frac{(t_1-t)2gh}{1+0.024\frac{l}{d}+\zeta_1}}\right).$$

Now since in ordinary ventilation the quantity $0.00367(t_1-\frac{1}{2}t)$ is always a small fraction, we have

$$Q=Q_1=0.268F\sqrt{\frac{(t_1-t)h}{1+\zeta\frac{l}{d}+\zeta_1}}\text{ cu. m.}$$

$$\left(=0.362F\sqrt{\frac{(t_1-t)h}{1+0.024\frac{l}{d}+\zeta_1}}\text{ cu. ft.}\right).$$

Inversely, the cross-section of conduit needed for the passage of Q cu. meters [cu. ft.] of air per second is

$$F=3.731Q\sqrt{\frac{1+\zeta\frac{l}{d}+\zeta_1}{(t_1-t)h}}\text{ sq. m.}\quad\left(F=2.76Q\sqrt{\frac{1+\zeta\frac{l}{d}+\zeta_1}{(t_1-t)h}}\text{ sq. ft.}\right).$$

It is evident that these calculations will become still more complex when there are changes of direction and cross-section in the conduit, or when the crosss-section F_1 of the orifice of efflux is different from the cross-section F of the whole conduit. At all events we may, as the variations of density here are small, employ the well-known coefficients and formulas of hydraulics, and we must accordingly assume the following (see Vol. I, sec. VI, ch. 4):

(1) For the flow of air through *an orifice in a thin partition*, for example through an open door, the coefficient of resistance is

$$\zeta_1=\left(\frac{F}{\alpha F_1}-1\right)^2.$$

where F is the cross-section of the conduit just back of the orifice, F_1 that of the orifice, and α the coefficient of contraction (0.6) of the current of air.

(2) For the entrance of the current of air into a *narrower conduit* we have $F_1=F$, and therefore the corresponding coefficient of resistance is, on an average,

$$\zeta_1=\left(\frac{1}{\alpha}-1\right)^2=\left(\frac{10}{6}-1\right)^2=\frac{4}{9}=0.444;$$

or we may place $\zeta=0.5$, as agreeing better with experience.

(3) For the entrance of the air into a larger conduit the coefficient of resistance is $\zeta_1=\left(\frac{F}{F_1}-1\right)^2$, where F is the cross-section of the larger and F_1 that of the smaller conduit. If v is the velocity of the air in the larger pipe and v_1 that in the smaller pipe, the head due to the corresponding resistance is

$$h_1=\zeta_1\frac{v^2}{2g}=\left(\frac{F}{F_1}-1\right)^2\frac{v^2}{2g}=\left(1-\frac{F_1}{F}\right)^2\frac{v_1^2}{2g},$$

and for small values of $\frac{F_1}{F}$ this may be written

$$h_1=\frac{v_1^2}{2g}.$$

(4) For the passage of a current of air through an elbow which forms a right angle the coefficient of resistance is

$$\zeta_1=1\ (\textit{nearly unity});$$

consequently the head due to the resistance is nearly equal to the head due to the velocity. For an elbow forming an *acute angle* ζ_1 is greater than unity, while for one that forms an *obtuse angle* it is less than unity.

(5) If the cross-section F_1 of the orifice of efflux of the air-conduit is different from the cross-section F of the latter—for example, if F_1 is the cross-section of a doorway at the end of the conduit—the velocity of efflux must be taken equal to

$$v_1 = 0.268\sqrt{\frac{(t_1-t)h}{1+\left(\zeta\frac{l}{d}+\zeta_1\right)\left(\frac{\alpha F_1}{F}\right)^2}}$$

$$\left(v_1 = 0.362\sqrt{\frac{(t_1-t)h}{1+\left(\zeta\frac{l}{d}+\zeta_1\right)\left(\frac{\alpha F_1}{F}\right)^2}}\right),$$

for the velocity in the conduit is $v=\frac{\alpha F_1}{F}v_1$, and the resistances in the conduit are proportional to the head

$$\frac{v_2}{2g}=\left(\frac{\alpha F_1}{F}\right)^2\frac{v_1^2}{2g}.$$

The quantity of air flowing through and discharging from conduit is

$$Q=\alpha F_1 v_1 = 0.268\alpha F_1\sqrt{\frac{(t_1-t)h}{1+\left(\zeta\frac{l}{d}+\zeta_1\right)\left(\frac{\alpha F_1}{F}\right)^2}}$$

$$\left(Q=0.362\alpha F_1\sqrt{\frac{(t_1-t)h}{1+\left(\zeta\frac{l}{d}+\zeta_1\right)\left(\frac{\alpha F_1}{F}\right)^2}}\right);$$

or if we substitute $d=\frac{4F}{p}$, where p is the periphery of the pipe, we get

$$Q=0.268\alpha F_1\sqrt{\frac{(t_1-t)h}{1+\left(\frac{1}{4}\zeta\frac{pl}{F}+\zeta_1\right)\left(\frac{\alpha F_1}{F}\right)^2}},$$

$$\left(Q=0.362\alpha F_1\sqrt{\frac{(t_1-t)h}{1+\left(\frac{1}{4}\zeta\frac{pl}{F}+\zeta_1\right)\left(\frac{\alpha F_1}{F}\right)^2}}\right).$$

Example.—An otherwise closed hall, ABC, Fig. 5, 6 m. [19.68 ft.] high, is connected with the outer air by a rectangular opening A, 0.15 m. [0.492 ft.] wide and 0.1 m. [0.328 ft.] high, and also by a vertical pipe CD, 0.15 m. [0.49 ft.] diameter and 12 m. [39.36 ft.] long; now if the average temperature in the hall is 20° C. [68° F.], the average temperature in the pipe

25° C. [77° F.], and the temperature of the outer air 10° C. [50° F.], what quantity of air will circulate through this hall per hour?

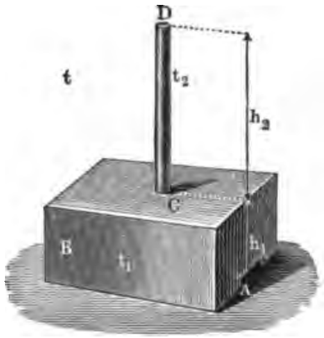

FIG. 5.

We must here make use of $(t_1 - t)h_1 + (t_2 - t)h_2$ instead of $(t_1-t)h$. In these expressions t_1 and h_1 represent the temperature and height of the hall, and t_2 and h_2 the temperature and height of the discharge-pipe. The expression therefore becomes $(20-10)6+(25-10)12=240$ [$=14.17$ for ft. and Fah. degrees]. Moreover, the area of the inlet is

$$F=15\times 10=150 \text{ sq. cm. } [0.161 \text{ sq. ft.}],$$

and that of the other pipe

$$F_1=\frac{\pi 15^2}{4}=176.7 \text{ sq. cm. } [0.19 \text{ sq. ft.}];$$

consequently the velocity of the entering air is

$$v=\frac{F_1}{F}v_1=\frac{176.7}{150}v_1=1.178v_1,$$

and the head due to this velocity is

$$\frac{v^2}{2g}=(1.178)^2\frac{v_1^2}{2g}=1.388\frac{v_1^2}{2g}.$$

Now if we regard the orifice as a short projection and assume for it the coefficient of resistance $\zeta_0=0.5$, the head required to force the air into the hall is

$$(1+\zeta_0)\frac{v^2}{2g}=1.5\times 1.388\frac{v_1^2}{2g}=2.08\frac{v_1^2}{2g}.$$

If we assume the coefficient of friction in the pipe to be $\zeta=0.032$, the head needed to remove the vitiated air is

$$\left(1+\zeta_0+\zeta\frac{l}{d}\right)\frac{v_1^2}{2g}=\left(1.5+0.032\frac{12}{0.15}\right)\frac{v_1^2}{2g}=4.06\frac{v_1^2}{2g},$$

and therefore the velocity with which the air passes through the pipe CD is

$$v_1=0.268\sqrt{\frac{240}{2.08\times4.06}}=1.675 \text{ m. [5.49 ft.]};$$

therefore the quantity flowing out of the hall per second is

$$Q_1=F_1v_1=0.01767\times1.675=0.0296 \text{ cu. m.};$$

hence the quantity per hour is

$$3600Q_1=106.5 \text{ cu. m. [3761. cu. ft.]}.$$

A person breathes per hour $\frac{1}{2}$ cu. m. [18 cu. ft.] of air, but in an inclosed space needs about 6 cu. m. [212 cu. ft.], and therefore this hall can properly accommodate $\frac{106.5}{6}=18$ persons.

§ 3. **Natural Ventilation.**—Natural ventilation in mines depends largely on the heat of the earth. The fluctuations of heat on the surface of the earth diminish as we go below the surface, and disappear entirely, in our climate, at a depth of about 24 m. [78.7 ft.]. From this point on, the heat of the earth increases very regularly at the rate of about 1° C. for every 30 m. [100 ft.] depth. The constant temperature at 24 m. [about 80 ft.] depth is on an average about 1° higher than the average temperature for the year at the surface, and for the middle of Germany and moderate elevation it is about $7°+1°=8°$ C. [46.4° F.]. During the year, in these localities the temperature at the surface varies between $-1°$ C. [30.2° F.] and $+17°$ C. [62.6° F.]; consequently the heat of the earth at a depth 300 m. [1000 ft.] is equal to the highest average temperature during the month of July. It follows that in mines with two openings A and D, as in Figs. 3 and 4, the air in winter will flow in at the lower opening A and out of the higher opening D (Fig. 3), for the air is then warmer inside of the pipes; on the other hand, in summer, when the outer air is warmer, the current will enter at the higher opening D and flow out of the lower one A (Fig. 4). But if both the orifices A and D are at least 24 m. [80 ft.] below the surface of the earth, the air-columns AB and CD will be in equilibrium, and artificial means will therefore be necessary to create a current

in either direction. Usually the temperature of the air is a few degrees higher than the surrounding rock, in consequence of the respiration of the workmen, the burning of lamps, etc., and this of course influences the draft. In order to assist or to generate ventilation in a mine, it is often necessary to bring it into the form of a conduit by means of brattices or partitions, or special air-passages may be built in the mine.

In drifts, such a brattice consists of an air-tight wooden partition, as *AB* in Fig. 6, and is used in connection with a cut-off *F* and an air-shaft *CD*.

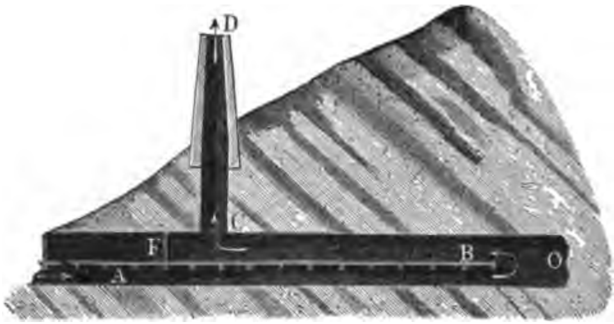

FIG. 6.

In winter the air moves under the brattice to the point *O*, and then above the brattice to, and out of, the shaft *CD*; in summer the air enters at *D* and traverses the mine in the opposite direction, *DCBA*.

The manner in which a mine may be ventilated by an *air-conduit* is shown in Fig. 7. *AB* is the shaft, *BC* a gallery, and *DEF* an air-conduit made of wood or sheet zinc. Usually the air in the conduit is the warmer and then the current passes down the shaft and out through the conduit. Instead of continuing the air-conduit beyond the shaft, a special hood is sometimes built over the shaft. Natural ventilation is more vigorous in winter when the air discharges at the higher points.

Example.—In order to furnish necessary ventilation for a drift *AO* (Fig. 6) 300 m. [984 ft.] long, 3 m. [9.84 ft.] high, and 1.2 m. [3.94 ft.] wide, a brattice is placed 1 m. [3.28 ft.] above the bottom of the drift, and at a distance of 30 m. [98 ft.] from the entrance to the drift is placed a square air-shaft *CD*, 20 m. [65.6 ft.] high and 1 m. [3.28 ft.] square. Required

the quantity of air that passes through the drift per second when the temperature outside is 13° C. [55.4° F.] and the average temperature of the air in the mine is 10° C. [50° F.].

FIG. 7.

The velocity with which the air flows through the air-shaft CD is given by the formula

$$v_1 = 0.268 \sqrt{\frac{(t_1 - t)h}{\left(1+\zeta_0+\zeta\frac{l}{d}\right)\left(\frac{F_1}{F}\right)^2 + 1 + \zeta_1 + \zeta\frac{l_1}{d_1}}}\,\text{m.}$$

$$\left(v_1 = 0.362 \sqrt{\frac{(t_1 - t)h}{\left(1+\zeta_0+\zeta\frac{l}{d}\right)\left(\frac{F_1}{F}\right)^2 + 1 + \zeta_1 + \zeta\frac{l_1}{d_1}}}\,\text{ft.}\right),$$

and in this we must substitute

$$(t_1 - t)h = (13 - 10)20 = 60 \quad [354].$$

Moreover, for the motion of the air in the shaft whose cross-section has the perimeter $p_1 = 4$ m. [13.12 ft.] and the area $F_1 = 1$ sq. m. [10.76 sq. ft.], we have

$$1 + \zeta_1 + \zeta\frac{l_1}{d_1} = 1 + 0.5 + 0.04\frac{p_1 l_1}{4F_1} = 1.50 + 0.80 = 2.30.$$

For the motion of the air under the brattice, since here $p = 2(1+1.2) = 4.4$ m. [14.43 ft.] and $F = 1.2$ sq. m. [12.92 sq. ft.], and $\zeta = 0.05$ on account of the roughness of the rock, we have

$$\left(1+\zeta_0+\zeta\frac{l}{d}\right)\left(\frac{F_1}{F}\right)^2=\left(1.5+0.05\frac{pl}{4F}\right)\left(\frac{F_1}{F}\right)^2$$
$$=\left(1.5+0.05\times\frac{4.4\times300}{4\times1.2}\right)\left(\frac{1}{1.2}\right)^2=10.59;$$

and for the return motion over the brattice, since $p=2(2+1.2)=6.4$, $F=2.4$ sq. m., and $l=300-30=270$ m. [886 ft.], we have

$$\left(1+\zeta_0+\zeta\frac{l}{d}\right)\left(\frac{F_1}{F}\right)^2=\left(1.50+0.05\times\frac{6.4\times270}{4\times2.4}\right)\left(\frac{1}{2.4}\right)^2=1.82,.$$

so that now

$$v_1=0.268\sqrt{\frac{60}{2.3+10.59+1.82}}=0.268\sqrt{\frac{60}{14.71}}=0.541 \text{ m.} \quad [1.77 \text{ ft.}],$$

and the quantity passing through the drift per second is

$$Q_1=F_1v_1=0.541 \text{ cu. m. } [19.1 \text{ cu. ft.}].$$

§ 4. **Artificial Ventilation.**—If the natural ventilation of a building or a mine does not suffice, *artificial ventilation* is employed, either by *heating* with *furnaces*, or by means of special *ventilating-machines*. In ventilating a building the stove is placed in the flue itself or outside of it, the latter being done when warming and ventilating are combined. The smoke of such a stove is always carried off by a pipe which passes up the middle of the flue. A simple ventilation with ventilating-flue is shown in Fig. 8. F represents an ordinary stove, intended to heat the space R; the air needed for combustion is furnished by the pipe A, and the products of combustion pass up the pipe EGH placed in the hot-air flue BCD. The vitiated air of the room enters the flue BCD at B, and while rising in the flue is heated by the smoke-pipe.

Another method of ventilation by heating the air is shown in Fig. 9. Here the furnace F is at a distance from the flue E, the products of combustion being discharged into the smoke-pipe GHK which passes up the flue. The furnace is surrounded by a metal drum within which the fresh air entering through the pipe A is heated. The air thus heated passes through the lateral openings in the drum into the room and from there through an opening C into the ventilating-flue E warmed

by the smoke-pipe. Instead of heating by a stove *BB*, the air may, of course, be heated by steam or hot water. Fig. 10

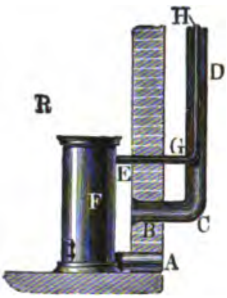

FIG. 8.

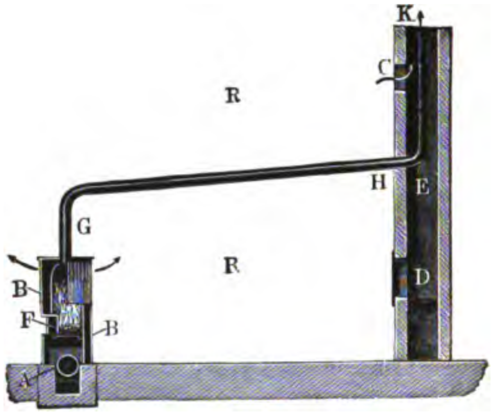

FIG. 9.

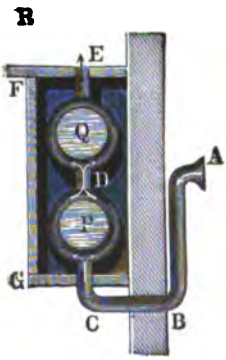

FIG. 10.

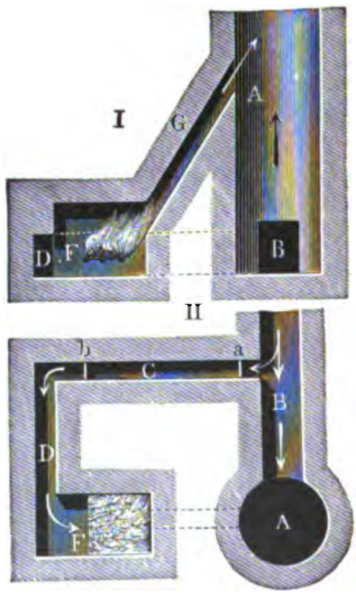

FIG. 11.

shows an arrangement for heating a room by these means. The pipes *P* and *Q* containing the warm water or steam are surrounded by tin pipes, and these are inclosed in a wooden box *FGD*. The air, admitted from without by the pipe *ABC*,

flows through the passage between the heated pipes and their envelopes, absorbs the requisite heat, and finally escapes through the pipe E into the space R to be heated. When larger spaces are to be heated, the furnace ABF, Fig. 9, is placed in a cellar and the smoke-pipe GHK is led off under the floor. In summer, when the space RR is not to be heated, the necessary ventilation can be obtained by a fire on the grate D in the ventilating-flue E.

The air-furnaces used in mines for ventilation are usually only short galleries driven from the air-shaft and provided with a grate. In order to utilize all the heat generated by the fire, the furnace must be placed as deep as possible. If the surrounding rock is not firm and dense, the furnace should be walled in. A vertical and a horizontal section of such a furnace are shown in Fig. 11, I and II. A small passage CD leads off from the main gangway B and passes to the fireplace F, and from this point rises in an air-passage G to the air-shaft A. Most of the air flowing through the main gangway passes directly into the air-shaft and is there heated by the products of combustion. The air-doors a and b in the side passage are provided with holes by means of which the furnace air can be regulated. To prevent explosions it is necessary to keep the air which is saturated with carbonic oxide gas from the fire, by discharging it into the shaft at a point at least 15 m. [50 ft.] from the discharging-orifice of the smoke-pipe G. For the same reason the air-furnace A, Fig. 12, is placed above or a little below the surface of the earth, and a flue C is set on the air-shaft B.

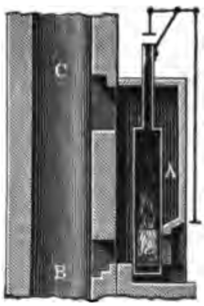

FIG. 12.

§ 5. **Theory of Artificial Ventilation.**—The velocities attending the ventilation of a mine $BCEH$, Fig. 13, by an *air-furnace* K and air-shaft EH can be expressed by special formulas deduced from the general formula in § 2. Let the difference of level AB between the outlets of the two shafts B and H be represented by h, the depth of the outlet F of the smoke-passage FK below the shaft-outlet B by $h_1 = BG$, and the perpendicular height of the smoke-passage by $h_2 = EF = CG$; moreover, let the temperature of the outer air $= t$, the average tem-

perature in $BCDEF$ be represented by t_1, the temperature in the upper part FH of the air-shaft by t_2, and the average temperature in the smoke-passage of the air-furnace by t_3; let us also designate the average cross-section of CDE by F_1, the cross-section of the air-shaft by F_2, and that of the smoke-

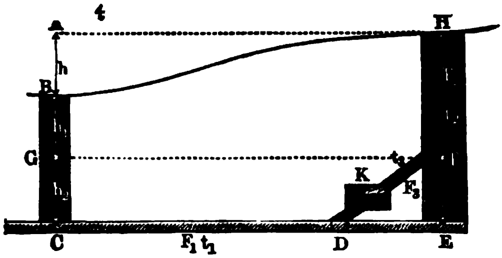

FIG. 13.

passage by F_3; finally, let the coefficient of resistance for the motion of the air in the mine CDE be represented by κ and that for the motion of the air through the smoke-passage by κ_1, then the velocity of the air flowing out at H is given by*

$$v_2 = 0.268\sqrt{\frac{(t_2-t)h+(t_2-t_1)h_1}{1+\kappa\left(\frac{F_2}{F_1}\right)^2}}$$

$$\left(v_2 = 0.362\sqrt{\frac{(t_2-t)h+(t_2-t_1)h_1}{1+\kappa\left(\frac{F_2}{F_1}\right)^2}}\right), \quad . \quad . \quad . \quad . \quad (1)$$

---

* $h(y-y_2)+h_1(y_1-y_2) = \frac{v_2^2}{2g}y_2 + \kappa\frac{v_1^2}{2g}y_1$, as $v_1 = \frac{F_2 v_2}{F_1}$;

$$h(y-y_2)+h_1(y_1-y_2) = \frac{v_2^2}{2g}y_2 + \kappa\left(\frac{F_2}{F_1}\right)^2\frac{v_2^2}{2g}y_1$$

$$= \frac{v_2^2}{2g}\left(y_2+\kappa\left(\frac{F_2}{F_1}\right)^2 y_1\right);$$

$$\therefore\ v_2 = \sqrt{2g\frac{(y-y_2)h+(y_1-y_2)h_1}{y_2+\kappa\left(\frac{F_2}{F_1}\right)^2 y_1}} = \sqrt{\frac{2g\left(\frac{y}{y_2}-1\right)h+\left(\frac{y_1}{y_2}-1\right)h_1}{1+\kappa\left(\frac{F_2}{F_1}\right)^2\frac{y_1}{y_2}}}.$$

Now $\frac{y}{y_2} = \frac{1+.00367t_2}{1+.00367t}\frac{p}{p_2}$; also $\frac{y_1}{y_2} = \frac{1+.00367t_2}{1+.00367t_1}\frac{p_1}{p_2}$; and as $\frac{p}{p_2}$ and

and consequently the volume discharged per second is

$$Q_2 = F_2 v_2. \quad \ldots \ldots \ldots \quad (2)$$

Furthermore, the velocity of the air in the mine CDE is

$$v_1 = \frac{F_2 v_2}{F_1}, \quad \ldots \ldots \ldots \quad (3)$$

and the velocity of the products of combustion at F, where they discharge into the air-shaft, is approximately

$$v_3 = 0.268\sqrt{\frac{(t_3 - t_1)h_2}{1+\kappa_1}} \quad \left(v_3 = 0.362\sqrt{\frac{(t_3 - t_1)h_2}{1+\kappa}}\right). \quad . \quad (4)$$

As the quantity of heat developed by the combustion is equal to that taken up by the air in the shaft, we have

$$F_1 v_1 (t_2 - t_1) = F_3 v_3 (t_3 - t_1),$$

and therefore

$$v_3 = \frac{F_1 v_1 (t_2 - t_1)}{F_3 (t_3 - t_1)}. \quad \ldots \ldots \ldots \quad (5)$$

Now if we equate the values given by (4) and (5) we obtain an equation for the determination of the temperature t_3 in

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$\frac{p_1}{p_2}$ may be taken equal to 1, we have

$$v_2 = \sqrt{\frac{2g\left(\frac{1+.00367t_2}{1+.00367t} - 1\right)h + \left(\frac{1+.00367t_2}{1+.00367t_1} - 1\right)h_1}{1+\kappa\left(\frac{F_2}{F_1}\right)^2\left(\frac{1+.00367t_2}{1+.00367t_1}\right)}} = \sqrt{2g}$$

$$= \sqrt{\frac{2g\frac{.00367(t_2 - t)h}{1+.00367t} + \frac{.00367(t_2 - t_1)h_1}{1+.00367t_1}}{1+\kappa\left(\frac{F_2}{F_1}\right)^2\left(\frac{1+.00367t_2}{1+.00367t_1}\right)}}.$$

Now assume that $1+.00367t$, $1+.00367t_1$, and $1+.00367t_2$ each $=1$, then we have approximately

$$v_2 = .0606\sqrt{2g\frac{(t_2 - t)h + (t_2 - t_1)h_1}{1+\kappa\left(\frac{F_2}{F_1}\right)^2}}.$$

the furnace, and then we can determine from either (4) or (5) the velocity v_3 and the quantity

$$Q_3 = F_3 v_3$$

passing through the air-chamber.*

Finally there remains to be determined the fuel needed for maintaining the fire in the air-furnace.

Let γ represent the specific weight of the air and w the quantity of heat developed by burning 1 kg..[1 lb.] of the fuel (see Vol. II); then, placing the specific heat of the air equal to 0.25, the required expenditure of fuel is found from

$$K = \frac{Q_3\gamma(t_3 - t_1)}{4w} = \frac{Q_2\gamma(t_2 - t_1)}{4w} \text{ kg. [lbs.].}$$

According to Vol. I, we have approximately

$$\gamma = \frac{1.702b}{1+0.00367t} \qquad \left(\gamma = \frac{0.0027b}{1+0.00204(t-32)}\right),$$

where b is the height of the barometer in meters [inches]; consequently for $b=0.760$ m. [29.92 ins.] and $t=0°$ C. [32° F.] we have $\gamma = 1.2935$ kg. [0.080 lbs. per cu. ft.].

If we substitute the value for v_2 given by (1) and that for Q given by (2) in the equation for K, we obtain

$$K = 0.268\frac{F_2\gamma(t_2 - t_1)}{4w}\sqrt{\frac{(t_2-t)h+(t_2-t_1)h_1}{1+\kappa\left(\frac{F_2}{F_1}\right)^2}}$$

$$\left(K = 0.362\frac{F_2\gamma(t_2 - t_1)}{4w}\sqrt{\frac{(t_2-t)h+(t_2-t_1)h_1}{1+\kappa\left(\frac{F_2}{F_1}\right)^2}}\right),$$

---

* $h_2(y_1 - y_3) = \frac{v_3^2}{2g}y_3 + \kappa\frac{v_1^2}{2g}y_1, \qquad v_1^2 = \left(\frac{F_3}{F_1}\right)^2 v_3^2;$

$$\therefore\; h_2(y_1 - y_3) = \frac{v_3^2}{2g}\left[y_3 + \kappa\left(\frac{F_3}{F_1}\right)^2 y_1\right].$$

$$v_3 = \sqrt{2g\frac{(y_1-y_3)h_2}{y_3+\kappa\left(\frac{F_3}{F_1}\right)^2 y_1}} = \sqrt{2g\frac{\left(\frac{y_1}{y_3}-1\right)h_2}{1+\kappa\left(\frac{F_3}{F_1}\right)^2\frac{y_1}{y_3}}} = \sqrt{2g\frac{\frac{.00367(t_3-t_1)h_2}{1+.00367t_1}}{1+\kappa\left(\frac{F_3}{F_1}\right)^2\frac{1+.00367t_3}{1+.00367t_1}}}.$$

$$v_3 = .0606\sqrt{2g\frac{(t_3-t_1)h_2}{1+\kappa\left(\frac{F_3}{F_1}\right)^2}} \quad \ldots\ldots (a)$$

t_3 can be obtained by equating (a) and (5).

and it follows from this that it is more economical to increase the draft by placing the air-furnace deeper than by raising the air to a higher temperature.

Example.—The outlet H of the air-shaft, Fig. 13, is at the distance $h=80$ m. [262.4 ft.] above the inlet B of the shaft through which the air enters; the outlet of the chimney of the furnace is at the distance $h_1=200$ m. [656 ft.] below B, and the height EF of the chimney or smoke-passage DF is $h_2=20$ m. [65.6 ft.]; the temperature of the outside air is $t=15°$ C. [59° F.], the average temperature of the air in the mine is $t_1=12°$ C. [53.6° F.] and that in the upper part of the air-shaft is $t_2=20°$ C. [68° F.]. The mine-passage has an average cross-section $F_1=3\times1.5=4.5$ sq. m. [48.5 sq. ft.], and a perimeter $p_1=2(3+1.5)=9$ m. [29.5 ft.], and its whole length $=l_1=3000$ m. [9840 ft.]; on the other hand the passage for the furnace has an average cross-section $F_3=1\times1.5=1.5$ sq. m. [16.1 sq. ft.], a perimeter $p_3=2(1+1.5)=5$ m. [16.4 ft.], and a length $l_3=$ 100 m. [328 ft.]. Finally the cross-section of the air-shaft $F_2=2\times3=6$ sq. m. [64.6 sq. ft.]. Required the quantity of air Q_2 which flows through the air-shaft per second, also the fuel needed. We have

$$(t_2-t)h+(t_2-t_1)h_1=(20-15)80+(20-12)200=2000 \quad [11,808],$$

also

$$\kappa\left(\frac{F_2}{F_1}\right)^2=\zeta\frac{p_1l_1}{4F_1}\left(\frac{F_2}{F_1}\right)^2=0.05\frac{9\times3000}{4\times4.5}\left(\frac{6}{4.5}\right)^2=133.3;$$

hence the velocity of the air leaving the air-shaft is

$$v_2=0.268\sqrt{\frac{2000}{1+133.3}}=1.034 \text{ m. [3.39 ft.] per sec.},$$

and the quantity of air discharged per second is

$$Q_2=F_2v_2=6\times1.034=6.204 \text{ cu. m. [28.2 cu. ft.]}.$$

Moreover, the average velocity with which the air traverses the gangway is

$$v_1=\frac{Q_2}{F_1}=\frac{6.204}{4.5}=1.379 \text{ m. [4.52 ft.] per sec.}$$

The velocity of the heated air is

$$v_s = 0.268\sqrt{\frac{(t_s - t_1)h_s}{1+\kappa_1}} = 0.268\sqrt{\frac{(t_s - 12)20}{1+.05\frac{p_s l_s}{4F_s}}}$$

$$= 0.268\sqrt{\frac{20(t_s - 12)}{1+0.05\frac{5\times100}{4\times1.5}}} = 0.527\sqrt{t_s - 12};$$

we also have

$$v_s = \frac{F_1 v_1 (t_2 - t_1)}{F_s(t_s - t_1)} = \frac{6.204\times8}{1.5(t_s - 12)} = \frac{33.08}{t_s - 12}.$$

From these two expressions for v_s we obtain

$$(t_s - 12)\sqrt{t_s - 12} = \frac{33.08}{0.527} = 62.77;$$

$$(t_s - 12)^{\frac{3}{2}} = 62.77;$$

$$t_s = 12 + \sqrt[3]{(62.77)^2} = 27.8^\circ \text{ C. } [82^\circ \text{ F.}]$$

This gives for the velocity of the heated air in the chimney

$$v_s = \frac{33.08}{t_s - 12} = \frac{33.08}{27.8 - 12} = 2.09 \text{ m. [6.86 ft.] per sec.};$$

hence the quantity of air heated per second is

$$Q_s = F_s v_s = 1.5\times2.09 = 3.135 \text{ cu. m. [110.7 cu. ft.]}.$$

If a kg. [lb.] of coal gives a useful heating effect of 5000 calories [9000 B.T.U.], the consumption of coal per hour is

$$3600K = 3600\frac{3.135\times1.2935\times15.8}{4\times5000} = 11.5 \text{ kg. [25.35 lbs.]}.$$

Remark.—According to Vol. II, the velocity of efflux of the air is given by the formula

$$v = \sqrt{\frac{\delta(t_1 - t)}{1+\delta t}2gh};$$

it will therefore be more exact to use here

$$v_2 = \sqrt{\frac{(t_2-t)h+(t_2-t_1)h_1}{1+\kappa\left(\frac{F_2}{F_1}\right)^2}2\delta g} = 0.268\sqrt{\frac{(t_2-t)h+(t_2-t_1)h_1}{1+\kappa\left(\frac{F_2}{F_1}\right)^2}},$$

$$v_2 = 0.268\sqrt{\frac{\frac{(t_2-t)h}{1+\delta t}+\frac{(t_2-t_1)h_1}{1+\delta t_1}}{1+\kappa\left(\frac{F_2}{F_1}\right)^2}}\left(v_2 = 0.362\sqrt{\frac{\frac{(t_2-t)h}{1+\delta t}+\frac{(t_2-t_1)h_1}{1+\delta t_1}}{1+\kappa\left(\frac{F_2}{F_1}\right)^2}}\right),$$

where $\delta = 0.00367$ [0.00208].

We should also reduce the quantity discharged to the outer temperature:

$$Q = \frac{1+\delta t}{1+\delta t_2}Q_2 = \frac{1+\delta t}{1+\delta t_2}F_2 v_2,$$

and should likewise take into account the variation of the density of the air with the temperature; for instance, instead of using

$$F_1 v_1 = F_2 v_2$$

we should employ

$$\frac{F_1 v_1}{1+\delta t_1} = \frac{F_2 v_2}{1+\delta t_2},$$

and instead of

$$F_2 v_2 (t_2 - t_1) = F_3 v_3 (t_3 - t_1)$$

the expression

$$\frac{F_2 v_2 (t_2 - t_1)}{1+\delta t_2} = \frac{F_3 v_3 (t_3 - t_1)}{1+\delta t_3}.$$

These corrections are insignificant for the ordinary values of t, t_1, t_2, t_3, and they have therefore been neglected above; for example, for $t = 15°$ C. [59° F.] and $t_1 = 12°$ C. [53.6° F.] we have

$$1+\delta t = 1+0.00367\times 15 = 1.055 \quad \text{and} \quad 1+\delta t_1 = 1.044,$$

and therefore in the last example we have for the efflux of the air

$$v_2 = 0.268\sqrt{\frac{\frac{400}{1.055}+\frac{1600}{1.044}}{1+133.3}} = 1.011 \text{ m. [3.32 ft.]},$$

instead of $v_2 = 1.034$ m. [3.29 ft.]: the air discharged reduced to the outer temperature now becomes

$$Q = \frac{1.055}{1.0734} \times 6 \times 1.011 = 5.96 \text{ cu. m. } [210.5 \text{ cu. ft.}].$$

§ 6. Double-acting Blowing-engines.—A vertical section of a large *double-acting blowing-cylinder* is shown in Fig. 14. This engine was built at Seraing, Belgium.* It consists of a single vertical blowing-cylinder CD, 1.83 m. [72.04 ins.] in diameter and 2.727 m. [8.94 ft.] high, and is driven directly by a steam-engine below it of 80 H.P. The diameter of the steam-cylinder is 1.05 m. [41.34 ins.], and the common stroke of the two machines is 2.44 m. [8 ft.]. In the figure KK is the blowing-piston and PR the piston-rod, 0.121 m. [4.76 ins.] in diameter. This rod is united by a sleeve coupling with the equally large piston-rod of the steam-engine, this coupling being a part of a cross-head 3.1 m. [10.17 ft.] long; this cross-head moves between two vertical guides, and is connected with two connecting-rods driving two fly-wheels, each 7.32 m. [24 ft.] in diameter and weighing about 90 cwt. V and V_1 are the suction-valves and W and W_1 are the valves admitting the air to the pipe WW_1L, which leads to the reservoir. The former are placed in pairs in the valve-chests, three of them being fastened to the top of the cylinder and three to the bottom. The rectangular valve-seats in these chests have an inclination of from 60° to 70° and cover valve-openings 0.5 m. [19.7 ins.] long and 0.25 m. [9.84 ins.] wide. On the other hand, the delivery-valves cover rectangular orifices 0.8 m. [31.5 ins.] long and 0.15 m. [5.9 ins.] wide. The valves are either of rubber or leather and, like the ordinary pump-valves, are stiffened by plates on both sides. This blower makes fourteen double strokes per minute and delivers air at a pressure of 250 mm. [9.8 ins.] of mercury, while the steam-engine works with a pressure of 3 atmospheres [44 lbs. per sq. in.] and one-third cut-off.

The *air-pumps* employed for creating a vacuum in closed spaces, for example in the *drive-pipes* of atmospheric railways, are not essentially different from blowers. Fig. 15 gives a section of such an air-pump used for the atmospheric railway

* See Portefeuille de John Cockerill, Pl. 31 to 34.

at St. Germain, near Paris. The air-cylinder is here 2.5 m. [98.4 ins.] in diameter and 2.2 m. [7.2 ft.] high, and has two suction-valves, V and V_1, and two delivery-valves, W and W_1, but in this machine the suction-valves communicate with

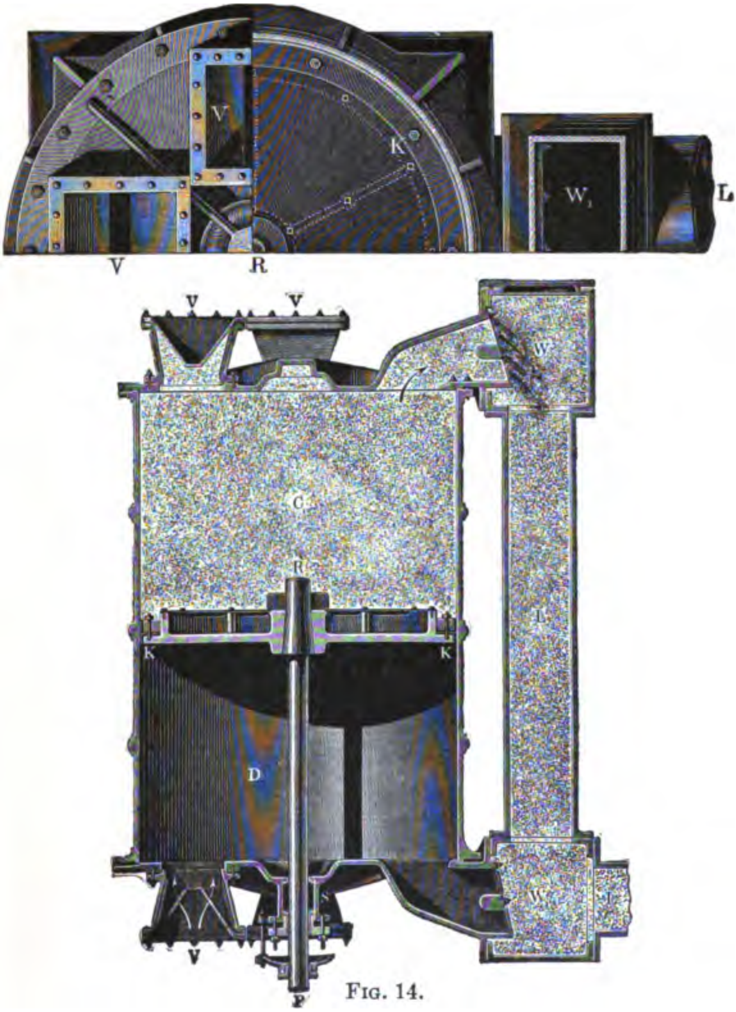

FIG. 14.

the drive-pipe by the channels A and B, while the delivery-valves communicate with the open air. These valves are made of bronze in order to be able to sustain a great pressure, and that they may be easily moved they are balanced

by arms VP, V_1P_1, WR, and W_1R_1, provided with counterweights. To prevent the suction-valves from opening too far, the arms have short tappets QQ_1, which, when the valves are open, strike against lugs in the valve-seat. In order that the delivery-valves W and W_1 may open and close without shock, they have little pistons FF_1 which suck air into

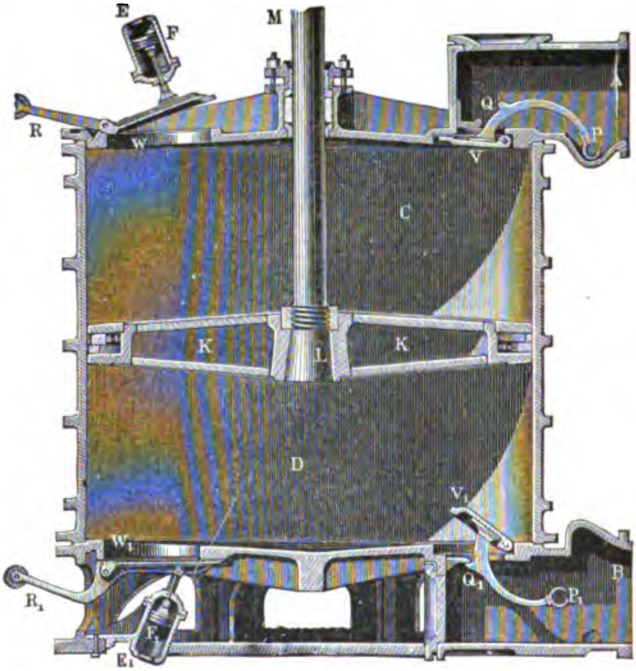

FIG. 15.

a small cylinder EF, E_1F_1 through small openings when the valves close, and drive out the air through these openings when the valves open. The metal-packed piston KK is driven by its rod ML, which in turn is driven by a crank-shaft. In order that the cylinder-blowers may work *rapidly*, large cross-sections must be provided for suction and delivery, but large valves of the requisite strength are very heavy and therefore open with difficulty; to obtain the same object it is therefore preferable to employ many small openings for the suction and delivery of air, and these are either separately covered by small leather or rubber valves, or the whole is covered by larger

valves of the same kind. The air-pump of an apparatus of this kind is shown in Fig. 16, and was used on the experimental railway at St. Quen.* The air-cylinder *CD* is here of sheet metal and is provided with a cast-iron top and bottom which

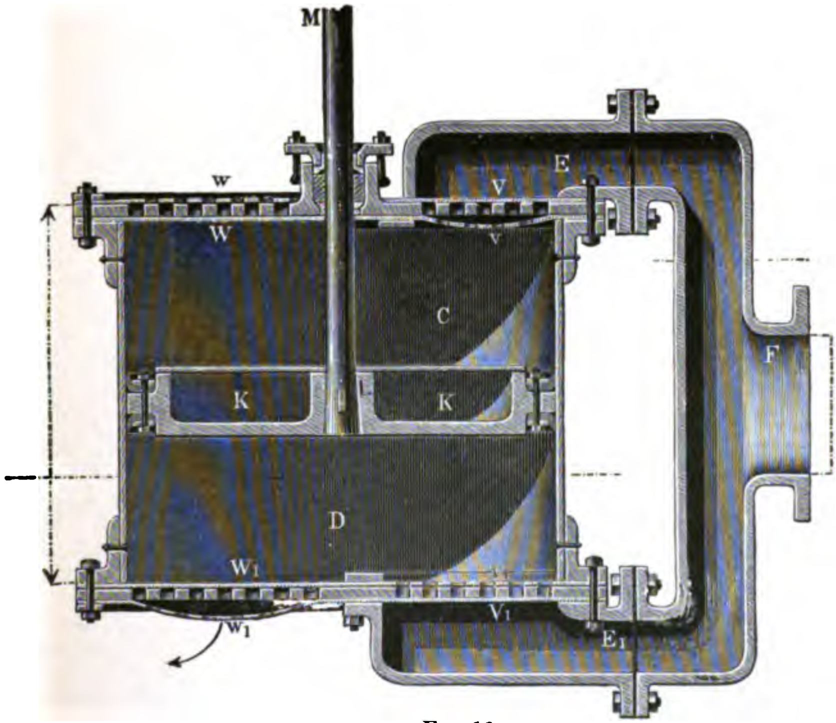

Fig. 16.

are perforated with many small holes 4 cm. [1.57 in.] in diameter. Half of these openings WW_1 communicate with the open air and the other half, VV_1, with the pipes EF, E_1F_1, opening into the space to be exhausted; the former are closed on the outside by the leather covers ww_1, and the latter by covers vv_1 of the same sort. These covers are fastened and made air-tight at their edges, but are also provided with many circular holes; these, however, do not coincide with the holes in the end of the cylinder, but fall between them. Now when the air is compressed on one side of the piston and rarefied on the other,

* See Armengaud, Publication industrielle, T. VI.

the elastic leather valves are lifted or pressed from their seats, so that the holes in the plate communicate with the holes in the leather and thus permit the air to flow in and out.

A horizontal blower is shown in section in Fig. 17.* The holes through which the air is sucked in here, lie on a large circle, as at VV, V_1V_1, and the holes connected with the blast-pipe lie in a smaller circle, WW, W_1W_1, near the piston-rod LM, the latter passing through stuffing-boxes in both cylinder-heads. The valves proper, or valve-clacks, are four rubber or leather rings fastened at their outer edges. The two larger

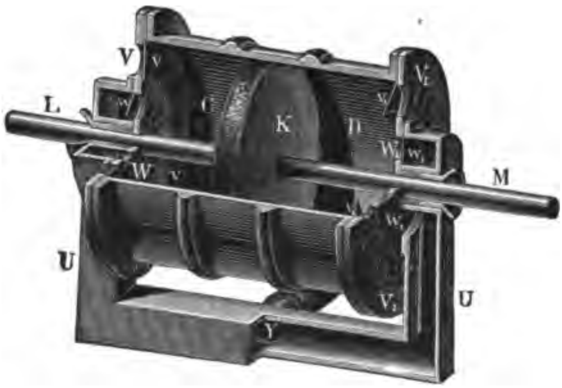

FIG. 17.

rings, vv and v_1v_1, cover the suction-orifices on the inside, and the two smaller ones, ww and w_1w_1, cover the blast-orifices on the outside. It is now easy to understand how the reciprocating motion of the piston KK sucks in air on one side through the outer circle of valves, and on the other side forces the air through the inner valve-openings into the blast-pipe.

Instead of arranging the suction- and blast-holes in a circle, they may be uniformly distributed over the whole cover, or at least arranged in straight lines so that the suction-orifices will occupy one half the cylinder-cover and the blast-orifices the other half. A vertical section of a part of a cover thus perforated is shown in Fig. 18, I and II, A, B, and C are the valve-passages, and da, db, dc their leather clack-valves covered with their metal plates and fastened at d, d, d.

* See Turner's Stabeisen- und Stahlbereitung, Bd. I.

In vertical blowing-cylinders simple lift-valves like those shown in Fig. 19, I and II, may be employed with advantage. These valves, *aa*, *bb*, *cc*, are provided with sleeves that slide over stationary pins *de*, *de*, and at the end of their lift strike against the disk-shaped heads *e*, *e*, *e*, of these pins. Valves have also been frequently made in the form of circular rubber

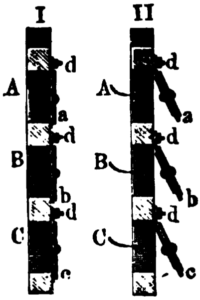

FIG. 18.

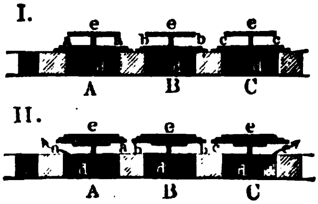

FIG. 19.

plates which rest on a star-shaped gridiron like that employed with corresponding pump-valves in Fig. 54, Mechanics of Pumping Machinery, an excessive lift being prevented by a suitably shaped stop-plate.

A horizontal blower with clack-valves is shown in vertical section in Fig. 20. It is of *Thomas & Laurent* design and was described in *Armengaud's* Publication industrielle, T. VIII. The blowing-cylinder CD has a diameter of 1.6 m. [63 ins.] and is fastened by four bolts to a cast-iron bed-plate. The upper halves of the cylinder-covers are provided with suitable suction-valves VV_1, and the lower valves with blast-valves WW_1; the latter communicate with the channels XX_1, and these with the blast-pipe Y. The cross-head has horizontal guides and is connected with the piston-rod of the steam-engine by a double yoke FF composed of four bars. To regulate the motion, a rod AE connects the cross-head A with a fly-wheel (not shown in the figure).

For modern American blowing-engines, see Appendix.

§ 7. **Tuyeres.**—The tuyere is a nozzle made of sheet iron soldered; it has a length of from 0.3 to 1.2 m. [12 to 48 ins.] and an orifice of from 2 to 10 cm. [0.74 to 4 ins.] diameter; the smaller ones are used in smith's forges, and the larger in blast-furnaces. The outlet of the tuyere does not extend to the fire

or smelting-chamber of the furnace, but is inside of a clay, iron, or copper bushing fitting the opening left for the admission of air to the furnace. This opening has the form of the frustum of a cone with either semi- or full circular bases. To prevent the bushings from melting, their walls are made hollow, a stream of cold water flowing through the space; they are called *water-tuyeres.* The position of the tuyere relatively to the smelting space depends upon the character of the smelting process; the axis of the bushing is nearly or quite horizontal, and its outlet

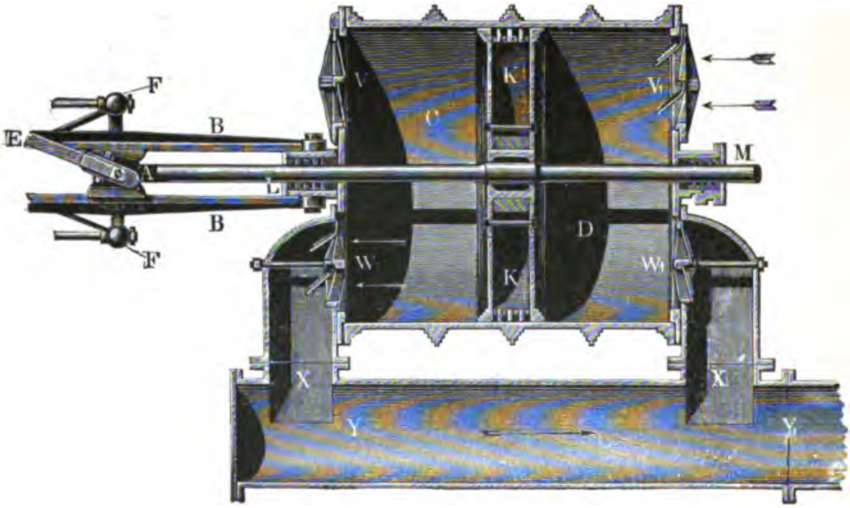

FIG. 20.

is either in the wall of the furnace or projects a little into the furnace space.

In order that the bushing may lead the air from the tuyere to the furnace without obstruction, it is necessary that the axis of the tuyere coincide as closely as possible with the nose of the bushing, and that the outlet of the tuyere and that of the bushing be nearly equal and as closely together as possible. Therefore tuyeres are not rigidly attached to the blast-pipe, but are connected with it by special devices which allow them to take any desired position. The simplest connection consists of a *leather hose* provided with iron-threaded rings, so that one end can be screwed to the blast-pipe and the other to the tuyere. Such an arrangement is shown in plan in Fig. 21. The blast-

pipe *WA* divides itself at *A* into three branches, *AC*, AE_1, and AE_2, which lead the air to the tuyeres *D*, K_1, and K_2; these

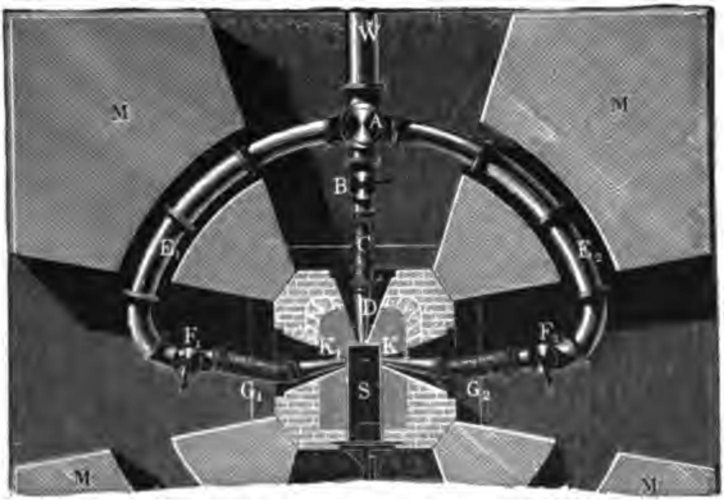

FIG. 21.

branch-pipes have regulating- or stop-valves, *B*, F_1, and F_2, and are also provided with the hose *C*, G_1, and G_2.

With a *hot* blast leather hose of course cannot be employed, and therefore a *fire-proof* mechanism is needed for adjusting the tuyeres. Such an apparatus is shown in Fig. 22. Here the

FIG. 22.

tuyere is provided with a special nozzle *CD* which has a ball-joint at *K*. The outside of *C* is turned and can be shifted in a stuffing-box *E*, placed at the end of an elbow in the blast-pipe *ABF*. To give the axis of the tuyere the right direction it is only necessary to turn the ball-joint; to move the tuyere back

and forth the screw *FG* is turned in the nut *F* attached to the tuyere. Finally, to regulate the supply of air a slide *S* is applied which moves in grooves and is also covered with a

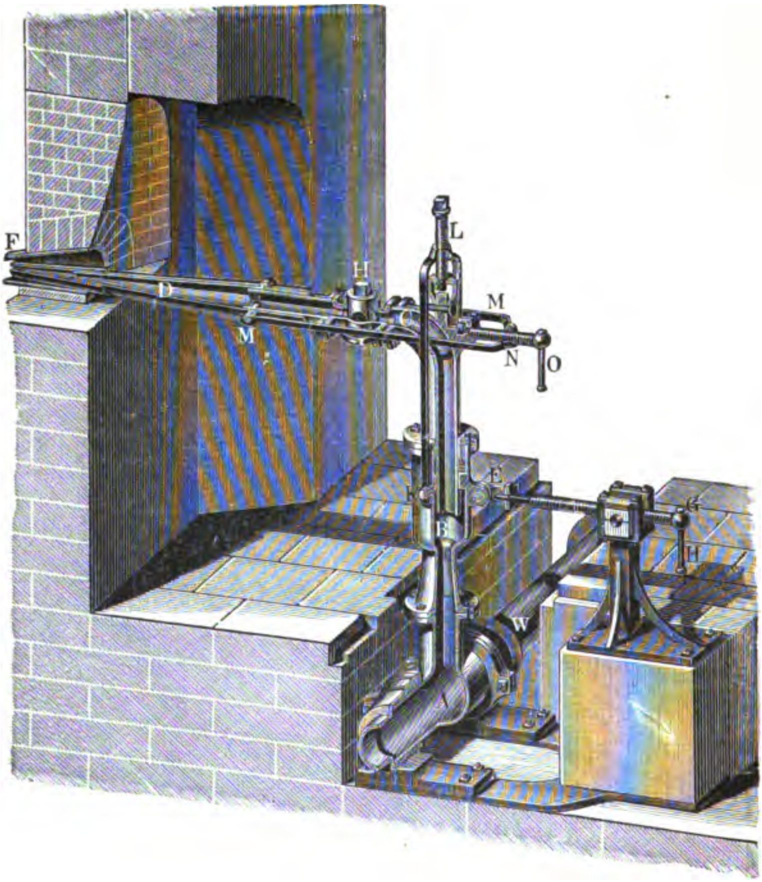

FIG. 23.

packing-ring; it is adjusted by a screw and crank mechanism *LMO*.

A more complete tuyere adjustment, such as is used in the blast-furnaces at *Freiberg*, is shown in Fig. 23.* Here the air-conduit *ABC* is so connected with the blast-pipe at *A* by two

* See von Herder's Schrift: Die vorzüglichsten Apparate zur Erwärmung der Gebläseluft, Freiberg, 1840.

stuffing-boxes that it can be moved forward and backward by the screw EG and crank GH, and in this way the tuyere D can receive any required position relatively to the horizon. Moreover, the upper part C of the conduit is so connected by a stuffing-box and rods KL with the lower part that it can be shifted in the latter by means of a screw L, and in this way the whole tuyere D can be set higher or lower. Besides, by turning the upper part of the conduit in the lower part the direction of the tuyere may be varied in a horizontal plane as desired.

Finally, the latter is pushed over the end of the pipe C and moved back and forth by the screw mechanism MNM. At F is shown the opening leading to the furnace, and at H the cock for regulating the blast.

§ 8. Hot-air Blast.—The use of hot-air blast has been found especially advantageous in making pig iron in blast-furnaces and melting it in cupolas, and in working iron in refining processes and in smith's fires. The heat contained in the hot blast produces a higher temperature in the furnace than is attainable with a cold blast, and when, as is usually the case, the heating is effected by furnace-gases which would otherwise be wasted considerable fuel is saved by applying the hot blast.

The blast is usually heated by passing the air compressed by the blowers and collected in the receivers, through a system of iron pipes subjected on the outside to the heat of the furnace-gases. The transmission of heat from the gases to the air passing through the pipes is to be discussed in the same way as the heating of water in steam-boilers: the transmission is more perfect the greater the surface of the system of pipes in comparison with their contents and the longer the air stays in the heating apparatus. At all events, the *surface* of the apparatus is to its *volume* as its *perimeter* p is to the *area* F *of the pipe*, and the heating period for a given velocity of flow of air in the pipes is directly proportional to their length l. The heating will therefore be better the greater the ratio $\frac{p}{F}$ and the length l of the pipes. The ratio $\frac{p}{F}$ is known to be smallest for circular cross-sections and greater for elliptical or rectangular cross-sections, and is also greater for two or more separated surfaces than for a single surface of the same cross-section. For exam-

ple, for a single cylindrical pipe of cross-section F and diameter $d=\sqrt{\frac{4F}{\pi}}$ the periphery is

$$p=\pi d=\sqrt{4F\pi},$$

while with n equal pipes which together have the same cross-section F the diameter is

$$d'=\sqrt{\frac{4F}{n\pi}}=\frac{d}{\sqrt{\pi}},$$

and the sum of the peripheries is

$$p'=n\pi d'=\sqrt{4n\pi F}=p\sqrt{n}.$$

Therefore it is advantageous in air-heating apparatus to employ a system instead of a single pipe, and it is well to choose an elliptical cross-section instead of a circular one, although in the former case the greater difficulty of construction and maintenance must be considered. As the frictional resistance of the air when passing through the pipes also increases with the value $\frac{p}{F}l$, due consideration must be given to this point in order that the loss of pressure of the air may not become disproportionately large. As iron pipes are liable to burn through and leak when greatly heated, a higher temperature of the hot blast than 300° C. [570° F.] is not allowable in them. On this account fire-bricks have recently been employed with advantage as a means of transmitting the heat of the furnace-gases to the blast. This is done by passing the gases through large receivers or chambers containing great masses of fire-brick. These masses are heated red-hot by the gases, and absorb, by virtue of their great specific heat (see Vol. II), a considerable quantity of heat. If, after the apparatus has been properly heated, the gases are shut off and the cold blast admitted, the air will be heated by contact with the red-hot masses and will pass to the furnace as a hot blast.

The action of this apparatus has therefore a certain resemblance to that occurring in *Siemens' regenerative furnace.*

Of course to keep the blast-furnace in constant action, several such arrangements are needed; there are generally

four to a blast-furnace, three of which are always heated by the gases, while the fourth is traversed by the blast.

By means of this apparatus invented by *Cowper* * and improved by *Whitwell* † the temperature of the hot blast can be brought up to 600° C. [1110° F.] and even higher.

The apparatus for heating the cold air is supplied with the requisite heat either by a special furnace or by the waste furnace-gases, as is done, for example, in cupolas, blast-furnaces, smelting-furnaces, reverberating-furnaces, etc. In the latter case the pipes are sometimes placed in the chimney or flue through which the products of combustion are discharged. In smith-fires the cold air is led through a system of passages in the cast-iron body surrounding the tuyeres before it is discharged through the tuyeres into the fire. In the larger heating apparatus, for example, such as have recently come into general use for iron-smelting furnaces, there are special heating chambers or hot-blast stoves which are usually heated by the gas discharged from the blast-furnaces, or, rarely, by coal fire. It is well to place the apparatus for heating the cold air as close to the tuyeres as possible, in order that the hot blast may cool as little as practicable. At first the hot-blast stove was placed on top of the blast-furnace, which required the cold blast to be led up to the stove and then the hot blast down to the tuyeres. Of late the furnace-gases are led down to the hot-blast stoves and high chimneys are employed to drive off the gases from the furnaces. In all hot-blast stoves the principle of *opposite currents* is employed, which consists in passing the cold blast through the stove in a direction *opposite* to that of the furnace-gases; this causes the coldest particles of air to first come into contact with those parts of the stove which are least heated, and, as the air grows hotter, it comes into contact with still hotter portions of the stove, with the final result that the air leaves the stove at a higher temperature than if both currents traversed it in the same direction; that is, it is higher than if the air left the stove at the places least heated by the departing gases.

A pipe apparatus for directly heating the blast was built

* Artisan, 1860, p. 275.

† See Zeitsch. deutsch. Ing., 1870, p. 402; 1875, p. 684; and 1877, p. 39.

for the iron-works at Hasslinghausen and is shown in Fig. 24 (see Zeitsch. deutsch. Ing., 1857). This apparatus consists of thirty-six elliptical pipes arranged in groups of six over each other, the ends being united so as to form a series of return coils.

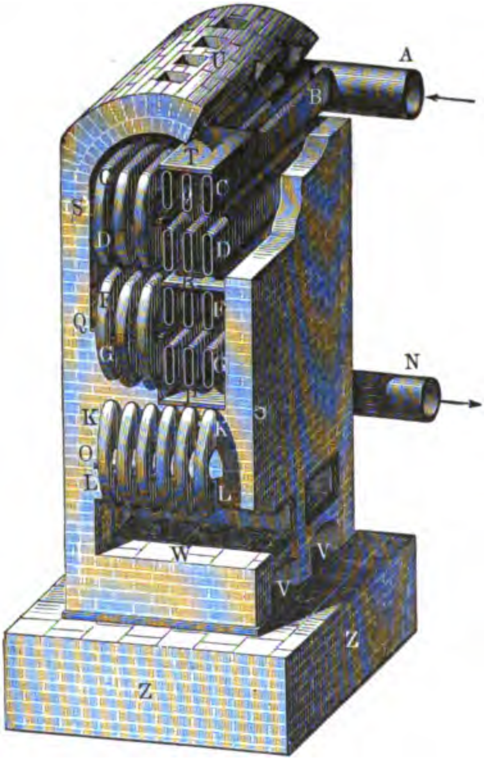

FIG. 24.

The upper ends B of these six series of pipes communicated with the pipe A supplying the cold blast, while the lower pieces L communicated with the pipe N which conducted away the hot blast. The gases generated by the fires on the two grates W pass through a series of return passages formed by the partitions O, P, Q, R, S, T, up to the outlets U, always touching the pipes on the outside and passing them in such a way that the cold blast first enters the upper and cooler pipes and gradually passes downward to the hotter pipes which are in contact with still hotter gases. The inside width of the pipes is 0.08 m.

and 0.47 m. [3.15 ins. and 18.75 ins.], and the length of the straight portion about 3 m. [9.84 ft.]. The elbows connecting these are placed outside the wall of the furnace so that they may be more easily repaired. The arrangement is not essentially different if, instead of heating with coal fire, the heating is effected with furnace-gases, for in this case it is only necessary to provide sufficient atmospheric air to burn completely the furnace-gases, consisting principally of carbonic oxide gas. To produce a strong current of the furnace-gases the outlets of all the hot-blast stoves are connected by a flue with a common high chimney. A hot-blast stove designed by Whitwell on the reverberation principle * is represented in vertical and horizontal section in Figs. 25 and 26. The apparatus is made

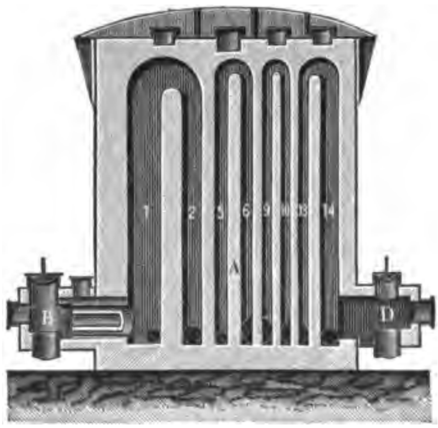

FIG. 25.

of fire-brick, is of cylindrical shape, and is inclosed by an iron envelope; its interior is divided by a central wall *a* and partitions at right angles to the latter, forming 14 spaces alternately connected above and below, so that the furnace-gases in traversing the stove from inlet *B* to outlet *D* pass through the cells in the order 1, 2, 3 . . . 14. There are slits in the partitions which communicate with the atmosphere and admit the air needed for the combustion of the gas; this insures uniform combustion in all the cells and a corresponding heating of the

* Revue universelle, 1869, Parts 5 & 6, and from this Zeitsch. deutsch. Ing., 1870, p. 402.

whole apparatus. The products of combustion are discharged through the outlet *D* which communicates with the passage leading to the chimney. After the apparatus has been sufficiently heated (ordinarily three hours are required with four stoves) the openings *B* and *B* for the furnace-gases are closed by suitable gear, the cold blast is now admitted at *C*, and after it has passed through the apparatus in the *opposite* direction, 14, 13, 12 . . . 1, escapes at *E* and passes to the tuyeres. The

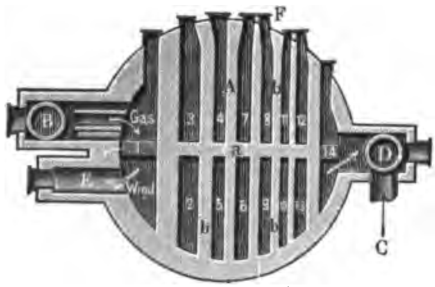

FIG. 26.

cold blast flows through the apparatus for about an hour, so that with four stoves, three always contain the furnace-gases and one the blast. In the four *Whitwell* stoves used at Geisweid the blast was changed every hour and the temperature of the hot blast at the beginning of the hour was found to be 600° C. [1110° F.] and at the end 550° C. [1020° F.]. By simultaneously employing two stoves these differences can be reduced still more. One defect of the *Whitwell* apparatus is that the soot from the furnace-gases deposits itself in the stove and when the cold blast is admitted it drives the soot and dust through the tuyeres into the furnace. To avoid this defect, the cleaning-doors *F* are provided. Moreover, the various cells gradually increase in volume from the inlet *C* to the outlet *E* of the blast, thus allowing for the expansion of the air as it is heated.

§ 9. **The Theoretic Work Performed by Blowers.**—The first and principal performance of an air-moving machine, whether a blower or an exhauster, consists of a compression of the air. If this were accompanied by no variations of the temperature, the compression would follow *Mariotte's law*, and the requisite quantity of work could be determined as follows:

1. Let the air in the blowing-cylinder C, Fig. 27, have the pressure p, and let the descending piston K force it into the reservoir R, which is already filled with air of a greater pressure, p_1. At first the piston traverses a certain part $AB=s_1$ of its whole stroke $AD=s$, before any air enters the reservoir R,

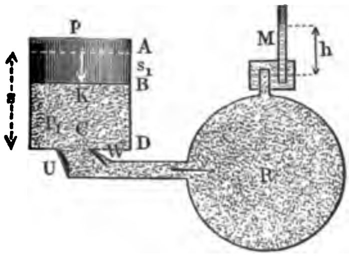

FIG. 27.

compression of the air from p to p_1 taking place instead. Let F be the piston area. Then the work needed for compression is (see Vol. I)

$$A_1=Fps\ \log_e \frac{p_1}{p}=Fps\ \log_e \frac{p_1}{p}.$$

While the piston traverses the rest of its stroke $s-s_1$, the resistance of the compressed air is constant, being Fp_1 for the whole piston area F; hence the corresponding expenditure of work is

$$A_2=Fp_1(s-s_1).$$

But the atmospheric air presses on the other side of the piston with a force Fp, and during the whole stroke performs the work $A_0=Fps$; consequently the whole work needed to press down the piston, neglecting all wasteful resistances, is

$$A=A_1+A_2-A_0=Fps\ \log_e \frac{p_1}{p}+Fp_1(s-s_1)-Fps;$$

or more simply, since *Mariotte's law* gives $ps=p_1(s-s_1)$, and hence $A_0=A_2$, we have

$$A=Fps\ \log_e \frac{p_1}{p}.$$

If we neglect all wasteful resistances on the return stroke and suppose the suction-valve V to have no weight, then during

this motion the forces on opposite sides of the piston K can be regarded as equal; it follows that the work performed on this stroke is equal to zero; hence the total work needed to compress the air from the density γ to γ_1, during which the pressure p becomes $p_1 = \frac{\gamma_1}{\gamma} p$, is

$$A = Fps \log_e \frac{p_1}{p} = Vp \log_e \frac{p_1}{p} = Vp \log_e \frac{b+h}{b}, \quad . \quad . \quad \text{(I)}$$

where $V = Fs$ is the volume of air forced out of the blowing-cylinder into the reservoir, b the height of the barometer, and h the height of the manometer on the reservoir R.

According to *Mariotte's law*,

$$(Fs)p = F(s-s_1)p_1, \quad \text{or} \quad Fps = Fp_1(s-s_1), \quad \text{or} \quad Vp = V_1 p_1,$$

where $V_1 = F(s-s_1)$ is the volume forced into the reservoir R, measured at the pressure in the latter, and hence

$$A = V_1 p_1 \log_e \frac{p_1}{p} = V_1 p_1 \log_e \frac{b+h}{b}. \quad . \quad . \quad . \quad . \quad (\text{I}_a)$$

2. Suppose the piston K, Fig. 28, to move up and down and to draw from the reservoir R, containing air at the lower

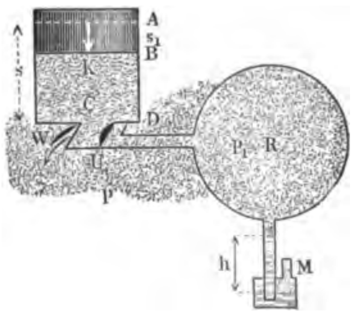

FIG. 28.

pressure, p_1, a certain volume of air $V_1 = Fs$ and to force it into the atmosphere which has the greater pressure, p. During the up-stroke of the piston the atmosphere opposes the motion of the piston with the force Fp, and the inner air assists the

motion with the force Fp_1; consequently the necessary expenditure of work is

$$A_1 = (Fp - Fp_1)s = Fs(p-p_1) = V_1(p-p_1).$$

To force this air of pressure p_1 from the blowing-cylinder into the outer air it must first be compressed by the piston and brought to the pressure p, which requires the work

$$A_2 = V_1 p_1 \log_e \frac{p}{p_1}.$$

After this compression, which changes the volume V_1 into $V = \frac{p_1}{p} V_1$, the delivery-valve W opens and the air presses with equal force on the two sides of the piston, and therefore no expenditure of work is needed for the remainder of the return-stroke.

If s_1 is the distance traversed by the piston during compression, the work with which the atmosphere assists compression is $A_0 = Fps_1$.

Now, according to *Mariotte's law*,

$$p(s-s_1) = p_1 s,$$

and hence

$$ps_1 = (p-p_1)s;$$

therefore

$$A_0 = Fs(p-p_1) = V_1(p-p_1) = A.$$

or the same as the opposing work, and the total work needed to move the volume V_1 of air from the reservoir into the atmosphere is

$$A = A_1 + A_2 - A_0 = A_2 = V_1 p_1 \log_e \frac{p}{p_1} = Vp \log_e \frac{p}{p_1}. \quad . \quad \text{(II)}$$

If h is the height of the manometer, *an excess of pressure of the atmosphere over the pressure in the reservoir*, we have

$$\frac{p}{p_1} = \frac{b}{b-h},$$

and therefore

$$A = Vp \log_e \frac{b}{b-h} = V_1 p_1 \log_e \frac{b}{b-h}. \quad . \quad . \quad . \quad (\text{II}_a)$$

From the agreement among the formulas (I), (I_a), (II), (II_a), we see that the total work needed to expand a certain volume of air is determined from the same formula as the work for compressing it, the difference being that the height h of the manometer is positive for compression and negative for expansion.

§ 10. The expressions developed in the preceding article for the work needed to compress and expand the air have sufficient accuracy for use in practice only when this variation in density is unaccompanied by any considerable variation of temperature; this constant temperature may be assumed when the difference of pressure p_1-p is small, for example less than $\frac{p}{20}$, or if the variation of density takes place so slowly that the heat developed in one case and absorbed in the other has time to put itself in equilibrium with the outer air. For the ordinary velocity of blowing-pistons we cannot assume such an equalization of temperature, but there are blowers, and particularly exhausters, in which the height h of the manometer, is less than 25 mm. [1 in.] of mercury, i.e., in which $\frac{p_1-p}{p}$ is but a trifle more than $\frac{1}{30}$; in such machines the formulas developed above are at once applicable. But when we have greater heights of the manometer or difference of pressure, the influence of heat on the variation of density is too great to be neglected, and therefore in determining the work we must make use of the formula

$$A=\frac{x}{x-1}\left(\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}-1\right)Vp, \quad . \quad . \quad . \quad . \quad \text{(III)}$$

developed in Vol. I, sec. VII, chap. 6; the quantity $x=1.42$ representing the ratio of the specific heat of air under constant pressure to that under constant volume. In most blowers, for example those used for blast for furnaces for iron-works, the excess of pressure of the blast does not exceed one-third of an atmosphere, i.e., $\frac{p_1}{p}$ is at most $\frac{4}{3}$; therefore, to apply the last expression to this case, we can develop it into a series

and retain only the first two or three terms, thus making the calculation simpler. We have

$$\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}=\left(1+\frac{p_1-p}{p}\right)^{\frac{x-1}{x}}=\left(1+\frac{h}{b}\right)^{\frac{x-1}{x}}$$

$$=1+\frac{x-1}{x}\frac{h}{b}+\frac{1}{2}\frac{x-1}{x}\left(\frac{x-1}{x}-1\right)\left(\frac{h}{b}\right)^2$$

$$+\frac{1}{6}\frac{x-1}{x}\left(\frac{x-1}{x}-1\right)\left(\frac{x-1}{x}-2\right)\left(\frac{h}{b}\right)^3;$$

hence

$$\frac{x}{x-1}\left(\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}-1\right)Vp=\left(1-\frac{1}{2x}\frac{h}{b}+\frac{x+1}{6x^2}\left(\frac{h}{b}\right)^2\right)\frac{Vh}{b}p,$$

so that now the above formula for the work per double stroke takes the form

$$A=\left(1-\frac{1}{2x}\frac{h}{b}+\frac{x+1}{6x^2}\left(\frac{h}{b}\right)^2\right)\frac{Vh}{b}p.$$

If γ represents the density of the fluid in the manometer, we have $p=b\gamma$, and the formula for the work becomes

$$A=\left(1-\frac{1}{2x}\frac{h}{b}+\frac{x+1}{6x^2}\left(\frac{h}{b}\right)^2\right)Vh\gamma. \quad . \quad . \quad . \quad (\mathrm{III}_a)$$

If we develop the first formula (I),

$$A=Vp\log_e\frac{p_1}{p}=Vp\log_e\frac{b+h}{b}=Vp\log_e\left(1+\frac{h}{b}\right),$$

into a series, we shall get (see first part of Vol. I)

$$A=\left(\frac{h}{b}-\frac{1}{2}\left(\frac{h}{b}\right)^2+\frac{1}{3}\left(\frac{h}{b}\right)^3\right)Vp=\left(1-\frac{1}{2}\frac{h}{b}+\frac{1}{3}\left(\frac{h}{b}\right)^2\right)Vp\frac{h}{b},$$

or

$$A=\left(1-\frac{1}{2}\frac{h}{b}+\frac{1}{3}\left(\frac{h}{b}\right)^2\right)Vh\gamma. \quad . \quad . \quad . \quad . \quad (\mathrm{IV})$$

If we were dealing with an incompressible fluid, as water, the expenditure of work needed to force the volume V of the fluid into a space in which the pressure was greater by the

amount $h\gamma$ than in the original space would be determined from the simple expression

$$A = Vh\gamma. \quad \ldots \ldots \ldots \quad \text{(V)}$$

Consequently, other things being equal, the work given by the first or heat formula is less than that given by the last or water formula by the amount

$$\Delta A = \left(\frac{1}{2x}\frac{h}{b} - \frac{x+1}{6x^2}\left(\frac{h}{b}\right)^2\right)Vh\gamma,$$

and the work deduced from *Mariotte's law* is less than that given by the water formula by the amount

$$\Delta A_1 = \left(\frac{1}{2}\frac{h}{b} - \frac{1}{3}\left(\frac{h}{b}\right)^2\right)Vh\gamma.$$

It is only for very small heights of the manometer, for example for $\frac{h}{b} = \frac{1}{100}$, that all three formulas give nearly the same value $Vh\gamma$ for the work, this being the same as that required for lifting and moving the water.

If we make $x = 1.42$, we get

$$\Delta A = \left(0.3521\frac{h}{b} - 0.2\left(\frac{h}{b}\right)^2\right)Vh\gamma;$$

for example, for $\frac{h}{b} = \frac{1}{20}$,

$$\Delta A = 0.0171Vh\gamma \quad \text{and} \quad \Delta A_1 = 0.0242Vh\gamma;$$

for $\frac{h}{b} = \frac{1}{10}$,

$$\Delta A = 0.0332Vh\gamma \quad \text{and} \quad \Delta A_1 = 0.0467Vh\gamma;$$

for $\frac{h}{b} = \frac{1}{5}$,

$$\Delta A = 0.0624Vh\gamma \quad \text{and} \quad \Delta A_1 = 0.0867Vh\gamma;$$

for $\frac{h}{b} = \frac{2}{5}$,

$$\Delta A = 0.1088Vh\gamma \quad \text{and} \quad \Delta A_1 = 0.1467Vh\gamma.$$

In the $\frac{2}{5}$ case the more exact heat formula gives

$$A=\frac{x}{x-1}\left(\frac{p_1}{p}^{\frac{x-1}{x}}-1\right)Vp=3.381\left(\left(\frac{7}{5}\right)^{0.2958}-1\right)\frac{5}{2}Vh\gamma=0.8847Vh\gamma.$$

Consequently $\Delta A=0.1153Vh\gamma$, and the formula based on *Mariotte's law* gives

$$A=Vp\log_e\frac{p_1}{p}=\frac{5}{2}Vh\gamma\log_e\frac{b+h}{b}=\frac{5}{2}Vh\gamma\log_e\left(\frac{7}{5}\right)=0.8412Vh\gamma;$$

hence

$$\Delta A_1=0.1588Vh\gamma.$$

We infer from this that for air-pressures where $\frac{h}{b}$ is greater than $\frac{1}{5}$ or h is more than 150 mm. [6 ins.] of mercury, the approximate formula,

$$A=\left(1-\frac{1}{2x}\frac{h}{b}+\frac{x+1}{6x^2}\left(\frac{h}{b}\right)^2\right)Vh\gamma,$$

for determining the work will no longer answer; hence we must use the fundamental formula

$$A=\frac{x}{x-1}\left(\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}-1\right)Vp=3.381\left(\left(\frac{b+h}{b}\right)^{0.2958}-1\right)Vp.$$

If the blower forces n cylinders of air into the reservoir per minute, the theoretical quantity of air delivered per second is

$$Q=\frac{n}{60}V=\frac{n}{60}Fs,$$

and the work required by this blower is, theoretically,

$$L=\frac{n}{60}A=3.381\left(\left(\frac{b+h}{b}\right)^{0.2958}-1\right)Qp,$$

or, approximately,

$$L=\left(1-0.3521\frac{h}{b}+0.2\left(\frac{h}{b}\right)^2\right)Qh\gamma.$$

Now if the whole blowing plant consists of n_1 single-acting pistons, each making n_2 double strokes per minute, then $n=n_1\times n_2$;

but if it consists of n_1 double-acting pistons which force air into the reservoir each stroke, then $n=2n_1\times n_2$, and for the first case we have

$$Q=\frac{nV}{60}=\frac{nFs}{60}=\frac{n_1 \cdot n_2}{60}Fs,$$

and for the second case

$$Q=\frac{n_1 \cdot n_2}{30}Fs.$$

If we are dealing with an exhauster which forces the quantity of air Q of the pressure $p_1=(b-h)\gamma$ into the atmosphere whose pressure is $p=b\gamma$, the necessary work is

$$L=3.381\left(1-\left(\frac{b-h}{b}\right)^{0.2958}\right)Qp,$$

or, approximately,

$$L=\left(1-0.3521\left(\frac{h}{b}\right)+0.2\left(\frac{h}{b}\right)^2\right)Qh\gamma,$$

as in blowers.

Example 1.—A blowing plant for a blast-furnace has two double-acting pistons, each 1.2 m. [47.24 ins.] in diameter, and a stroke of 1 m. [3.28 ft.]; each piston makes ten double strokes per minute and produces a blast of 900 mm. [35.48 ins.] pressure from atmospheric air of 750 mm. [29.53 ins.] pressure. Required the theoretical work needed per second.

Here $n_1=2$ and $n_2=10$; moreover,

$$F=\frac{\pi d^2}{4}=1.131 \text{ sq. m. [1753 sq. in.]},$$

and $s=1$ m. [3.28 ft.]; hence the volume of blast delivered per second is

$$Q=\frac{n_1 \cdot n_2}{30}Fs=\frac{2\times 10}{30}1.131=0.754 \text{ cu. m. [26.63 cu. ft.]}.$$

Now, $h=0.900-0.750=0.150$ m. [5.9 ins.], and the specific gravity of mercury is $\gamma=13.6$; hence, if the air be regarded as an incompressible fluid, the required work becomes

$$Qh\gamma=0.754\times 0.15\times 13{,}600=1538.2 \text{ m.-kg.}=20.51 \text{ H.P.}$$

But, on account of the increase of pressure connected with the variation of density, the required theoretical work is

$$L=\left(1-0.3521\frac{h}{b}+0.2\left(\frac{h}{b}\right)^2\right)Qh\gamma$$

$$=\left(1-0.3521\frac{150}{750}+0.2\left(\frac{15}{75}\right)^2\right)1538.2$$

$$=0.9376\times1538.2=1442.2 \text{ m.-kg.}=19.23 \text{ H.P.}$$

Example 2.—An exhausting-machine has two single-acting pistons, each 3 m. [9.84 ft.] in diameter and with 2 m. [6.56 ft.] stroke; each makes twelve double strokes per minute and works against an excess of pressure on the side of the atmosphere of 0.1 m. [0.328 ft.] of water. Required the theoretical quantity of air exhausted by this machine per second, and also the theoretical work needed for this purpose. Here

$$F=\frac{\pi d^2}{4}=7.069 \text{ sq. m. [76.1 sq. ft.]};$$

consequently the quantity of air per second is

$$Q=\frac{n_1 \cdot n_2}{60}Fs=\frac{2\times12}{60}7.069\times2=5.655 \text{ cu. m. [200 cu. ft.]}.$$

Now, if the mercury barometer stands at 760 mm. [29.53 ins.], the water barometer stands at 10.2 m. [33.5 ft.], and we have $\frac{h}{b}=\frac{0.1}{10.2}=0.0098$, and as $h=0.1$ m. [0.33 ft.] and $\gamma=1000$ kg. (about 62.43 lbs. per cu. ft.], we can place $h\gamma=100$ [20.5]; hence the work required is

$$L=\left(1-0.3521\frac{h}{b}\right)Qh\gamma=(1-0.00345)5.655\times100$$

$$=563.5 \text{ m.-kg.}=7.51 \text{ H.P.}$$

§ 11. **The Clearance or Hurtful Space.**—In consequence of the hurtful space and the imperfect motion of the valves, the blowers and exhausters take up a volume V of air per stroke which is *less* than the space swept through by the piston; therefore, judging of the performance of such a blower, we must take into account the loss of air. Let F again represent the piston

area AB, Fig. 29, σ_1 the height $DG=CE$ of the clearance or hurtful space in the cylinder, F_2 the area, and l_2 the length of the hurtful space in the valve-space L. Then the quantity of air remaining in the blower AEL when the piston has completed its stroke $AD=BC=s$ is

$$V_1=F\sigma_1+F_2l_2;$$

or if, as in Vol. II, we make this equal to $F\sigma$, the height of y cylinder having the same area as the piston and a volume equal to the total hurtful space is

$$\sigma=\frac{V_1}{F}=\sigma_1+\frac{F_2l_2}{F}.$$

As the remaining quantity of air $F\sigma$ has the inner pressure p_1, the suction-valve M will not at once open when the piston begins its return-stroke, for the inclosed air must first expand till it has fallen to the pressure p of the outer air. The distance $CH=DK=\lambda$ traversed by the piston during this time is determined by *Mariotte's law*,

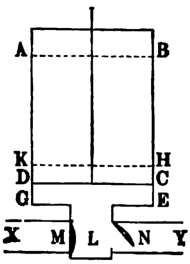

FIG. 29.

$$\frac{\sigma+\lambda}{\sigma}=\frac{p_1}{p},$$

and therefore

$$\lambda=\left(\frac{p_1}{p}-1\right)\sigma=\left(\frac{b+h}{b}-1\right)\sigma=\frac{h}{b}\sigma.$$

For the remainder of the stroke $s-\lambda$ the quantity of air

$$F(s-\lambda)=F\left(s-\frac{h}{b}\sigma\right)=Fs\left(1-\frac{h}{b}\frac{\sigma}{s}\right)$$

flows into the blowing-cylinder; therefore, in consequence of the expansion of the air in the hurtful space, the loss of air is $W=\frac{h}{b}\frac{\sigma}{s}Fs$, and its ratio to the quantity of air is $\frac{W}{V}=\frac{h}{b}\cdot\frac{\sigma}{s}$.

We found above for the theoretical quantity of air delivered by a blower per second

$$Q=\frac{n}{60}Fs;$$

accordingly the effect of the actual quantity measured under the outer pressure p is

$$Q_1=\frac{n}{60}Fs\left(1-\frac{h}{b}\frac{\sigma}{s}\right).$$

Hence the loss of blast due to the hurtful space increases with the height σ of this space and with the excess of pressure or manometer height h. In blowers of low pressure, for instance where $\frac{h}{b}=\frac{1}{20}$, this loss is insignificant, for if $\frac{\sigma}{s}$ were as much as $\frac{1}{10}$, this loss would still be only $\frac{1}{20}\times\frac{1}{10}=\frac{1}{200}$, i.e., $\frac{1}{2}$ per cent. of the theoretical volume of the air.

While the piston is traversing the distance $CH=DK=\lambda$ the imprisoned air presses with a greater force upon the piston than the outer air, and therefore the piston not only needs no force to drive it, but receives from the expanding air in the hurtful space the quantity of work

$$(F\sigma)p_1 \log_e \frac{p_1}{p}=F\sigma p_1 \log_e \frac{b+h}{b},$$

so that the necessary quantity of work per double stroke is

$$\begin{aligned}A&=F(s+\sigma)p \log_e \frac{p_1}{p}-F\sigma p_1 \log_e \frac{p_1}{p}\\&=[Fsp-F\sigma(p_1-p)] \log_e \frac{p_1}{p}=Fp\left(s-\frac{h}{b}\sigma\right)\log_e \frac{p_1}{p}\\&=Fp(s-\lambda) \log_e \frac{b+h}{b},\end{aligned}$$

which is exactly equal theoretically to that required for delivering the quantity of air $V=F(s-\lambda)$.

Note.—In a blower without clearance the work in a double stroke is only that of compression. In a blower with clearance the work is like that of a blower without clearance minus work of expansion in clearance space; i.e., in a blower with clearance the total work is equal to work of compression minus work of expansion.

Similar relations obtain with the exhauster. Here the hurtful space is filled with atmospheric air when the piston is in its lowest position CD, and the distance $CH=DK=\lambda$

traversed by the piston while the suction-valve remains closed is given by the formula

$$\frac{\sigma+\lambda}{\sigma}=\frac{b}{b-h},$$

so that we get

$$\lambda=\left(\frac{b}{b-h}-1\right)\sigma=\frac{h}{b-h}\sigma.$$

Here the ratio of the loss of air to the theoretical quantity is

$$\frac{W}{V}=\frac{\lambda}{s}=\frac{h}{b-h}\frac{\sigma}{s},$$

and the actual quantity measured at the inner pressure p_1 is

$$Q_1=\frac{n}{60}Fs\frac{(s-\lambda)}{s}=\frac{n}{60}Fs\left(1-\frac{h}{b-h}\frac{\sigma}{s}\right).$$

As $\frac{h}{b-h}$ is greater than $\frac{h}{b}$, this loss of volume is greater in exhausters than in blowers, other things being equal. For

$$\frac{h}{b-h}\frac{\sigma}{s}=1, \text{ i.e., for } h=\frac{s}{s+\sigma}b, \quad \left\{\begin{array}{l} h\sigma=(b-h)s, \\ h(\sigma+s)=bs, \end{array}\right\}$$

$$\text{or} \qquad \sigma=\frac{b-h}{h}s,$$

we even have $Q_1=0$, because under these circumstances the pressure of the air imprisoned in the hurtful spaces does not equal

$$\frac{\sigma}{s+\sigma}b=\frac{\sigma}{s}h=b-h$$

till the piston has reached the end of its stroke; consequently the suction-valve M does not open at all. But this loss of air is again accompanied by a gain of work, for the piston at the beginning of the up-stroke need not exert the whole force $Fh\gamma$, but exerts it only after the piston has traversed the distance $\lambda=\frac{h}{b-h}\sigma$.

Hence the work needed by the exhauster per double stroke is given by

$$A = Fp_1(s-\lambda) \log_e \frac{p_1}{p} = Fp_1\left(s - \frac{h}{b-h}\sigma\right) \log_e \left(\frac{b}{b-h}\right).$$

Now although the preceding developments were based on *Mariotte's law*, the results are sufficiently accurate in practice because here it is only a question of determining a small correction.

For this reason we can write as above, for the work required by the blower per second,

$$L = \left(1 - 0.3521\frac{h}{b} + 0.2\left(\frac{h}{b}\right)^2\right)Qh\gamma,$$

where Q is the quantity of air moved per second, the general formula for which is

$$Q = \frac{nF}{60}(s-\lambda),$$

the special formula for blowers being

$$Q = \frac{nFs}{60}\left(1 - \frac{h}{b}\frac{\sigma}{s}\right),$$

and the special formula for exhausters

$$Q = \frac{nFs}{60}\left(1 - \frac{h}{(b-h)}\frac{\sigma}{s}\right);$$

but we must not forget that in the first case the volume of air is measured at *atmospheric pressure* (b) and in the second case at the *inner pressure* ($b-h$).

§ 12. **Losses Due to the Clack-valves.**—A loss of work worth noticing occurs from the valves not opening until subjected to a certain excess of pressure, so that the pressure of air in a blowing-cylinder is less during suction than the pressure of the outer air, while during the forcing action the pressure is greater than that in the reservoir or blast-pipe. Let G_1 represent the weight of the suction-valve D_1E_1, Fig. 30, b_1 its lever-arm D_1H_1, G_2 the weight of the blast-valve D_2E_2, and b_2 its lever-

arm D_2H_2; then the moment of the opening force of one valve must be G_1b_1 and that of the other G_2b_2.

Let F_1 be the area of the suction-opening, F_2 of the blast-

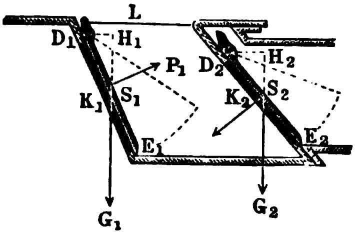

FIG. 30.

opening, a_1 the distance D_1K_1 of the middle K of the first orifice from the axis D, and a_2 that of the second orifice. Then the height z_1 of a fluid column having the density γ, which measures the excess of pressure in the first case, may be found from

$$F_1z_1\gamma a_1 = G_1b_1;$$

hence

$$z_1 = \frac{G_1b_1}{F_1a_1\gamma},$$

and in the second case

$$F_2z_2\gamma a_2 = G_2b_2;$$

hence

$$z_2 = \frac{G_2b_2}{F_2a_2\gamma}.$$

Therefore during suction the pressure in the blowing-cylinder is not $b\gamma$ but $(b-z_1)\gamma$, and it is easy to see that these excesses are attended by a loss of air and also by a loss of work. To reduce these losses as much as practicable the valves must be as light and the lever-arms of their weights as small as possible. For the last reason they should not be placed horizontally, but suspended in an inclined position.

The quantity of air taken in during an up-stroke, neglecting the loss due to the hurtful space, is $V = Fs$, and its pressure is $(b-z_1)\gamma$; it follows that the quantity of air reduced to the atmospheric pressure $b\gamma$ is

$$V_1 = \frac{b-z_1}{b}V = \left(1-\frac{z_1}{b}\right)Fs;$$

hence the loss of blast in consequence of the excess of pressure z_1 is

$$V-V_1=\frac{z_1}{b}Fs=\frac{z_1}{b}V.$$

The force with which the air in the cylinder tends to push up the piston is $F(b-z_1)\gamma$, and the work performed by it is $A_1=Fs(b-z_1)\gamma$.

During compression the pressure passes from $b-z_1$ to $b+h+z_2$, and the work during compression is

$$A_2=Fs(b-z_1)\gamma\ \log_e\frac{b+h+z_2}{b-z_1},$$

and the work done during the following forcing action is

$$A_3=F(s-s_1)(b+h+z_2)\gamma.$$

Now, according to *Mariotte's law,*

$$(s-s_1)(b+h+z_2)=s(b-z_1);$$

hence we have

$$A_1=A_3.$$

Therefore the whole work during a piston-stroke is

$$A=A_2+A_3-A_1=Fs(b-z_1)\gamma\log_e\frac{b+h+z_2}{b-z_1},$$

for the volume

$$V_1=\left(1-\frac{z_1}{b}\right)Fs$$

has its pressure changed from b to $b+h$.

But we have approximately

$$\frac{b+h+z_2}{b-z_1}=\frac{b+h}{b}+\frac{z_1+z_2}{b},$$

and if we neglect a few quantities and for a small value of x assume $\log_e (1+x)=x$, we get

$$A=Fsb\gamma \log_e \left(\frac{b+h}{b}+\frac{z_1+z_2}{b}\right)-Fsz_1\gamma \log_e \frac{b+h}{b}$$

$$=Fsb\gamma\left(\log_e \left(\frac{b+h}{b}\right)+\frac{z_1+z_2}{b+h}\right)-Fsz_1\gamma\frac{h}{b};$$

that is,

$$A=Fsb\gamma \log_e \frac{b+h}{b}+Fs\gamma\left(\frac{b}{b+h}(z_1+z_2)-\frac{h}{b}z_1\right),$$

or, if we also neglect $\frac{h}{b}$ in the last term, we get

$$A=Fsb\gamma \log_e \frac{b+h}{b}+Fs(z_1+z_2)\gamma$$

$$=Vp \log_e \frac{b+h}{b}+V(z_1+z_2)\gamma.$$

Exactly similar relations are found for the exhauster. While the air is being drawn from the reservoir into the cylinder, the pressure in the latter is $b-h-z_1$; consequently the quantity of air per stroke reduced to the pressure $b-h$ in the reservoir is

$$V_1=\frac{b-h-z_1}{b-h}Fs,$$

and the loss of air is

$$V-V_1=\frac{z_1}{b-h}Fs=\frac{z_1}{b-h}V.$$

The force with which the air tends to push the piston up or forward is $F(b-h-z_1)\gamma$, and the corresponding work is

$$A_1=Fs(b-h-z_1)\gamma.$$

On the other hand, during the return-stroke of the piston compression takes place from $b-h-z_1$ to $b+z_2$, and the work of compression is

$$A_2=Fs(b-h-z_1)\gamma \log_e \frac{b+z_2}{b-h-z_1},$$

and the work needed to force the cylinder air into the atmosphere is

$$A_3 = F(s-s_1)(b+z_2)\gamma.$$

But we have

$$(s-s_1)(b+z_2) = s(b-h-z_1);$$

hence $A_3 = A_1$, and therefore the total work per double stroke is

$$A = A_2 + A_3 - A_1 = A_2 = Fs(b-h-z_1)\gamma \log_e \frac{b+z_2}{b-h-z_1}.$$

If we again make

$$\frac{b+z_2}{b-h-z_1} = \frac{b}{b-h} + \frac{z_1+z_2}{b-h}, \text{ etc.,}$$

we have

$$A = Fs(b-h)\gamma \log_e \frac{b}{b-h} + Fs\gamma \frac{b-h}{b}(z_1+z_2),$$

or, if we neglect $\frac{h}{b}$ in the last term,

$$A = V(b-h)\gamma \log_e \frac{b}{b-h} + V(z_1+z_2)\gamma$$

$$= Vp_1 \log_e \frac{b}{b-h} + V(z_1+z_2)\gamma.$$

Therefore the loss of work due to the resistance of the valves in *exhausters* as well as in *blowers* is

$$\varDelta A = V(z_1+z_2)\gamma = V\left(\frac{G_1 b_1}{F_1 a_1} + \frac{G_2 b_2}{F_2 a_2}\right).$$

If we express the useful work by the more exact formula, we obtain for the *work of a blower per second*

$$L = \left\{\left(1 - 0.3521\frac{h}{b} + 0.2\left(\frac{h}{b}\right)^2\right)h\gamma + \frac{G_1 b_1}{F_1 a_1} + \frac{G_2 b_2}{F_2 a_2}\right\}Q.$$

Example.—Let us suppose the barometer to stand at $b = 750$ mm. [29.53 ins.], the height of a manometer of a blower

to be $h=80$ mm. [3.15 ins.], the weight of the valve per square meter of valve-orifice to be

$$\frac{G_1}{F_1}=\frac{G_2}{F_2}=50 \text{ kg [10.24 lbs. per sq. ft.]},$$

and the lever-arm of this weight to be one-fourth of the valve width, that is, $\frac{b_1}{a_1}=\frac{b_2}{a_2}=\frac{1}{4}$; hence the work needed per second is

$$L=\left\{\left(1-0.3521\frac{80}{750}+0.2\left(\frac{80}{750}\right)^2\right)0.08\times 13{,}600+50\times\frac{1}{4}+50\times\frac{1}{4}\right\}Q$$
$$=(0.9648\times 1088+25)Q=1074.7Q \text{ m.-kg. } [L=220Q \text{ ft.-lbs.}].$$

If the piston area $F=1.2$ sq. m. [18.92 sq. ft.] and the average piston velocity $v=0.9$ m. [2.95 ft.], the theoretical quantity of air measured at the atmospheric pressure is

$$Q=Fv=1.2\times 0.9=1.08 \text{ cu. m.};$$

therefore the loss of volume of the blast due to the weight of the valve is

$$\Delta Q=\frac{z_1}{b}Q=\frac{50Q}{4b\gamma}=\frac{50}{4\times 0.750\times 13{,}600}Q=0.00122Q=0.0012 \text{ cu. m.}$$
$$=[0.043 \text{ cu. ft.}],$$

which is very insignificant, and finally the performance is

$$L=1074.7\times 1.08=1160.7 \text{ m.-kg. [horse-power].}$$

Other and greater losses of blast occur in *piston-blowers* with *clack-valves*, due to the escape of the air when the valves are closing, and to leakage of the valves after they are closed. Of course while the suction-valve is closing a certain quantity of air escapes from the cylinder into the open air, and while the delivery-valve is closing a certain quantity comes back from the reservoir into the cylinder; moreover, as the blower-valves are not perfectly air-tight, some air flows from the reservoir into the cylinder while suction is going on, and while the air is being forced into the reservoir a part of the blast in the cylinder escapes into the open air. Hence in these blowers we must

estimate the actual quantity of blast per second, measured at atmospheric pressure, to be

$$Q_1 = xQ = 0.6Q \text{ to } 0.7Q.$$

We see from the above that the hurtful space is responsible for but a small part of this loss. In ordinary piston-valves with clacks this loss is of no consequence whatever, for the loss of work which it occasions is simply that needed to overcome the hurtful resistances, principally the piston friction, while the piston is traversing the distance λ, and this distance according to *Mariotte's law* $= \frac{h}{b}\sigma$, and according to *Poisson's law* is only equal to $\frac{h}{xb}\sigma = 0.704\frac{h}{b}\sigma$.

§ 13. **Losses Due to the Slide-valves.**—The influence of the hurtful space on the action of the blower is very different in *piston-blowers* with *slide-valves*. In order to get a clear insight into the action of such a blower, we will suppose its piston K to occupy the three successive positions I, II, and III, Fig. 31. The piston in these three positions is near the end of its stroke, and the valve is near the middle of its stroke. In position I the piston is traveling from left to right, in position II it is at the end of its stroke, and in III it occupies the same position as at I, but is on its return-stroke moving from right to left, while the valve in all these positions is moving from right to left. At the instant when the piston reaches I the passages M and N in the blowing-cylinder are closed by the valve, thus stopping suction on the left end and also stopping the forcing of air into the reservoir R from the right end. While the piston passes from position I to II both cylinder-passages are closed and the piston K traverses the last portion $KL = s_1$ of its stroke

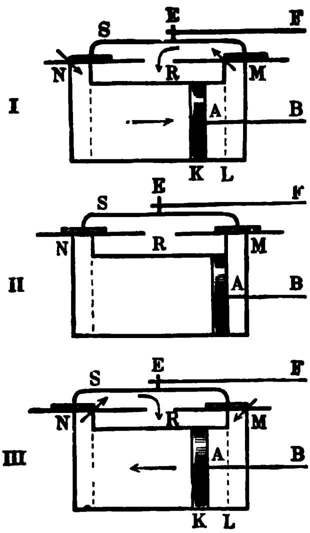

FIG. 31.

without drawing in or forcing out air. This closure of cylinder-passages continues till the valve has passed from position II to III and the piston has again reached the first position I, whereupon the air imprisoned on both sides returns to the first condition of density. Supposing the valve to now continue its motion from right to left, both passages will open and the piston K will also move from right to left, forcing the air on the left side through N into the reservoir and filling the right side with fresh air drawn through M. This forcing and suction process is greatest when the piston is at the middle of its stroke and the cylinder-passages are fully open. Let F again represent the piston-area, s the piston-stroke, and σ the reduced height of the hurtful space; then for position III the volume at the left of the piston is

$$C_1=F(s-s_1+\sigma),$$

and that at the right, reduced to the atmospheric pressure, is

$$C_2=F(s_1+\sigma)\frac{b+h}{b}.$$

When the air-passages are next closed the valve occupies the same position as in I, while the piston is as near the opposite end of the stroke as it is from the beginning of the stroke in III; consequently the volume of air at the left of the cylinder, reduced to atmospheric pressure, is

$$C_2=F(s_1+\sigma)\frac{b+h}{b},$$

and the volume at the right of the piston is

$$C_1=F(s-s_1+\sigma);$$

hence the quantity of air drawn in and forced out per stroke is

$$V_1=C_1-C_2=F\left(s-s_1+\sigma-(s_1+\sigma)\frac{b+h}{b}\right),$$

i.e.,

$$V_1=F\left(s-2s_1-(s_1+\sigma)\frac{h}{b}\right),$$

and the corresponding loss of blast is

$$V-V_1=F\left(2s_1+(s_1+\sigma)\frac{h}{b}\right).$$

The volume of air $F(s_1+\sigma)$ imprisoned by M at position III escapes into the air as soon as the valve moves on, the pressure of this air changing from $b+h$ to b; in so doing a volume of blast is lost equal to

$$\Delta V=F(s_1+\sigma)\frac{b+h}{b}-F(s_1+\sigma)=F\frac{h}{b}(s_1+\sigma).$$

There is still another irregularity in a blower with slide-valves, namely, when the valve moves from position III into the following position the air-passages are opened, and a part of the blast rushes back from the reservoir into the cylinder. The work needed by a blower with slide-valves to generate the volume of blast

$$F\left(s-2s_1-(s_1+\sigma)\frac{h}{b}\right)$$

is therefore

$$A=\left(1-0.3521\frac{h}{b}+0.2\left(\frac{h}{b}\right)^2\right)F(s-2s_1)h\gamma.$$

The loss of air just found and the return-flow of the blast from the reservoir to the cylinder can be reduced by setting the valve so that it will not occupy its middle position when the piston is at the end of its stroke, the valve being set so that it will be a little back of the piston motion and the air-passages M and N being opened by it at the instants when the pressures of the air in the cylinder are equal to the pressures in the blast-pipe and suction-pipe respectively. The manner of doing this will now be shown.

§ 14. **The Indicating of Blowers.**—The determination of the pressures of the air in the blowing-cylinder during a piston-stroke may be made by placing an indicator on it and drawing a curve whose ordinates represent the pressure of the air for every piston position.

Two such indicator-curves are shown in Fig. 32, I and II. They were taken from a large double-acting *piston-blower* with

clack-valves, the diameter being 2.2 m. [86.6 ins.] and the stroke 1.6 m. [63 ins.]. The piston-curve I corresponds to 17, and the curve II to 13, double strokes per minute. The lower part, *ABCD*, of such a diagram is drawn by the indicator-pencil during suction, and the upper part, *DEFA*, during the forcing. We

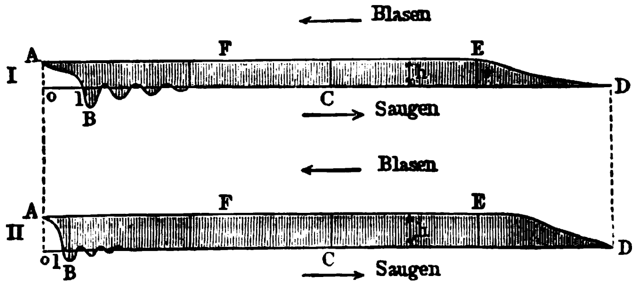

FIG. 32.

see that the suction-valve does not open immediately at the beginning of a stroke, but after the piston has traversed a part *oI* of its path *oD*, and that the valve makes a few oscillations before it is completely open. With rapid piston speed (I) the suction-valve is not continually open until the piston has traversed one-fourth of its stroke, but with the slow motion (II) it is constantly open after one-sixth of its stroke. While the piston is traversing the remainder of its stroke, the pressure of the air in the cylinder is very constant and very nearly that of the atmosphere, for the corresponding line *CD* of the indicator-curve almost coincides with the zero-line *oD*. The upper part, *DEFA*, of the curve is drawn during compression and blowing; we see from the curve that in the first case (I) the piston traverses a quarter, and in the second case nearly a sixth, of a stroke before the air assumes a constant pressure (h). Moreover, the pressure is greater in the second case than in the first, which is explained by the fact that in the former the quantity of air generated with the same working force is smaller.

In Fig. 33, I and II, two other indicator-diagrams are given, which were taken from a blower with slide-valves, the diameter being 1.34 m. [52.76 ins.] and the stroke 1 m. [39.37 ins.]. The first curve (I) was obtained when the number of double strokes was 70 per minute, the average pressure of the blast $h = 1.5$ cm. [0.59 in.] of mercury, and on an average the valve was 5.8 cm.

[2.28 ins.] behind the piston; on the other hand, when the second curve was taken the number of double strokes per minute was sixty, the average pressure of the blast 1.45 cm. [0.57 ins.] of mercury, and on an average the valve was only 9 mm. [0.35 in.] behind the piston. Therefore when the piston is at the end of its stroke the valve must in the first case travel 58 mm. [2.28 ins.] and in the second case 9 mm. [0.35 ins.] before the air-passages open. The lower part, *ABCD*, of such a curve is here also drawn during the suction part of the piston motion, while the upper part, *DEFA*, belongs to the compressing and blowing part of the motion. We see that in the second case

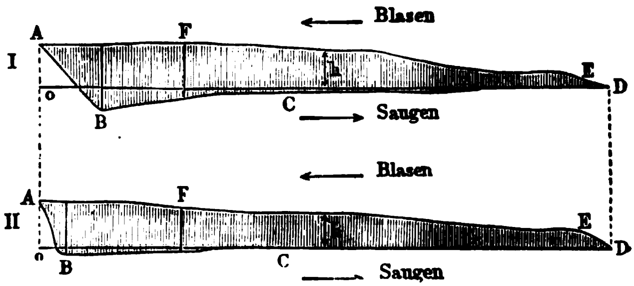

FIG. 33.

(II) suction takes place much earlier in the stroke than in the first case (I), and that the vacuum produced is smaller and lasts a shorter time than in I. We also see, from the course of the upper curve, that compression in II is more regular and greater than in I, but still much more variable than in blowers with *clack*-valves (see Fig. 32, II), where a pretty constant pressure occurs soon after the piston begins its stroke. But a comparison of curves I and II in Fig. 33 shows that a greater speed and a greater setting back of the valve reduce the efficiency of the blower (see Publication industr. par M. Armengaud ainé, Vol. XII).

§ 15. **Resistances in Blowers with Clack-valves.** — The work theoretically necessary in a cylinder blower, whose piston area is F and stroke s, according to Vol. I, is given by

$$\frac{x}{x+1}\left(\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}-1\right)pFs=A_0, \quad . \quad . \quad . \quad . \quad (1)$$

where $p=b\gamma$ represents the atmospheric pressure and $p_1=(b+h)\gamma$, the pressure in the blast reservoir. But additional work is needed to overcome certain hurtful resistances, for instance the resistances experienced by the air passing through the valve-openings and through the pipe connecting the blower with the reservoir, and also piston friction.

Let p' represent the air-pressure on that side (A) of the piston K at which suction takes place; then, according to § 12, the excess of pressure $p-p'=z_1\gamma$ of the atmosphere over the pressure of the air in the cylinder is given by

$$z_1\gamma=\frac{G_1b_1}{F_1a_1},$$

where F_1 is the valve area, G_1 the weight of the clack, b_1 its lever-arm, and a_1 the half-width of the valve. In consequence of this excess of pressure the piston must perform the work

$$A_1=z_1\gamma Fs=\frac{G_1b_1}{F_1a_1}Fs. \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Moreover, the pressure just back of the delivery-valve F_2 is greater than the pressure p_1 in the reservoir by the amount

$$\rho=\zeta\frac{l}{d_2}\frac{v_2^2}{2g}\gamma_1,$$

this excess being needed to overcome the frictional resistance in the connecting pipe. In the formula l is the length, d_2 the diameter of the connecting pipe, v_2 the average velocity of the air in it, and γ_1 the density of this air. As the delivery-valve has an area F_2, a weight G_2 with lever-arm b_2, and the half-width a_2, there must exist in the cylinder, on the forcing side B of the piston, an excess of pressure $z_2\gamma_1$ which is needed to keep the valve open and which, according to § 12, is given by

$$z_2\gamma_1=\frac{G_2b_2}{F_2a_2}.$$

The total excess of pressure on the side B of the piston over the pressure p_1 in the reservoir is therefore given by

$$z_2\gamma_1+\rho=\frac{G_2b_2}{F_2a_2}+\zeta\frac{l}{d_2}\frac{v_2^2}{2g}\gamma_1,$$

so that the resistance caused by the delivery-valve and the connecting pipe is

$$A_2 = \left(\frac{G_2 b_2}{F_2 a_2} + \zeta \frac{l}{d_3} \frac{v_3^2}{2g} \gamma_1\right) Fs. \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Now if we write the usual expression $4\phi \frac{e}{d} k$ for the piston friction, where k is the piston force, d the piston diameter, e the width of the packing, and ϕ the coefficient of friction, we finally get for the total work required for a single stroke of the double-acting blower the expression

$$A = \left(1 + 4\phi\frac{e}{d}\right)(A_0 + A_1 + A_2) =$$

$$\left(1+4\phi\frac{e}{d}\right)\left\{\frac{x}{x+1}\left(\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}} - 1\right)p + \frac{G_1 b_1}{F_1 a_1} + \frac{G_2 b_2}{F_2 a_2} + \zeta \frac{l}{d_3}\frac{v_3^2}{2g}\gamma_1\right\} Fs, \quad (4)$$

in which we must substitute $\frac{p_1}{p} = \frac{b+h}{b}$ and $v_3 = \frac{F}{F_3} v$, where F_3 is the cross-section of the pipe connecting the blower with the reservoir. If the blower delivers n cylinder volumes of air to the reservoir per minute, the requisite work per second is

$$L = \frac{n}{60} A. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

We must here remark that the preceding determination of the heads z_1 and z_2 measuring the excess of pressure back of the clack-valves was based on the supposition that the valve-openings were sufficiently large to prevent the losses of head due to the passage of air through these openings from becoming greater than the heads z_1 and z_2 due to the valve-weights G_1 and G_2

Let F_m be the smallest cross-section of the suction-valve passage; when the air has passed through this passage its maximum velocity v_m suddenly changes to a smaller velocity v_n, and, according to Vol. I, this is accompanied by a loss of head expressed by $\frac{(v_m - v_n)^2}{2g}$. Therefore if α is the coefficient

of contraction for the opening F_m of the suction-valve,* we have for this valve

$$\frac{(v_m - v_n)^2}{2g} = \frac{(v_m - v)^2}{2g} = \left(\frac{v_m}{v} - 1\right)^2 \frac{v^2}{2g} = \left(\frac{F}{\alpha F_m} - 1\right)^2 \frac{v^2}{2g},$$

the velocity v_n of the air back of this valve being placed equal to the velocity v of the piston.

If we suppose the atmospheric air back of the suction-valve to be at rest and, after passing through the valve-opening, to follow the piston with a velocity v, there will be needed to accomplish this an excess of pressure z_1' of the atmosphere over the pressure in the suction space of the cylinder, which is determined by

$$z_1' = \left(\left(\frac{F}{\alpha F_m} - 1\right)^2 + 1\right)\frac{v^2}{2g}.$$

Now if the value z_1' is greater than

$$z_1 = \frac{G_1 b_1}{F_1 a_1 \gamma},$$

then this value z_1' must be employed in the formula for determining the work A, while z_1 is to be employed whenever z_1' is smaller than z_1. In this case the valve will open far enough to make its orifice

$$F_m = \frac{F}{\alpha\left(\sqrt{\frac{G_1 b_1}{F_1 a_1 \gamma}\frac{2g}{v^2} - 1} + 1\right)},$$

which expression can be obtained by equating the values of z_1 and z_1'.

In like manner we obtain for the delivery with opening F_m'' the loss of head due to passage through the valve, namely

$$\frac{(v_m - v_n)^2}{2g} = \frac{(v_m - v_3)^2}{2g} = \left(\frac{v_m}{v_3} - 1\right)^2 \frac{v_3^2}{2g} = \left(\frac{F_3}{\alpha F_m''} - 1\right)^2 \frac{v_3^2}{2g}$$
$$= \left(\frac{F_3}{\alpha F_m''} - 1\right)^2 \left(\frac{F}{F_3}\right)^2 \frac{v^2}{2g}.$$

* This opening F_m is to be distinguished from the cross-section F of the suction-valve, the quantity F_m depending on the amount the valve is open, and the cross-section F_1 being regarded as the maximum value of F_m.

Let us suppose the air back of the delivery-valve to move with the velocity v, and after it has passed through the valve to move with the velocity $v_3 = \frac{F}{F_3} v$ in the communicating pipe; then we need for this purpose an excess of pressure z_2' in the cylinder, which may be determined from

$$z_2' + \frac{v^2}{2g} = \left(\left(\frac{F_3}{\alpha F_m''} - 1 \right)^2 + 1 \right) \left(\frac{F}{F_3} \right)^2 \frac{v^2}{2g}.$$

The remarks made in connection with z_1 and z_1' are also applicable to z_2 and z_2'; here also the greater of the values z_2 and z_2' must be employed in determining the work A.

Example.—A cylinder-blower contains two double-acting cylinders, each 1.5 m. [4.92 ft.] in diameter, and each piston has a stroke of 1.6 m. [5.25 ft.] and makes fifteen double strokes per minute. Required the work needed when, with a height of barometer $v = 0.760$ m. [29.92 ins.], there is produced a blast with an effective pressure of 0.160 m. [6.30 ins.] of mercury.

Here the theoretical quantity of air for each single piston-stroke is

$$V = Fs = \frac{\pi \times 1.5^2}{4} \times 1.6 = 2.827 \text{ cu. m. [99.84 cu. ft.]},$$

and the theoretical work per second is

$$A_0 = \frac{x}{x-1} \left(\left(\frac{b+h}{b} \right)^{\frac{x-1}{x}} - 1 \right) pV$$

$$= 3.381 \left(\left(\frac{.920}{.760} \right)^{0.2958} - 1 \right) 0.760 \times 13{,}600 V$$

$$= 2031.6 V \text{ m.-kg. [416} V \text{ ft.-lbs. when } V \text{ is in cu. ft.]}$$

If the ratio of the lever-arms of the valves is $\frac{b_1}{a_1} = \frac{b_2}{a_2} = \frac{1}{2}$ and the weight of the valves per square meter is $\frac{G_1}{F_1} = \frac{G_2}{F_2} = 100$ kg. [20.48 lbs. per sq. ft.], then the work needed for the passage through the suction-valve is, according to (II),

$$A_1 = \frac{G_1 b_1}{F_1 a_1} Fs = \frac{1}{2} \times 100 \times V = 50 V \text{ m.-kg. [10.24} V \text{ ft.-lbs.]}$$

If the ratio of the length l of the connecting pipe to its average diameter is $\frac{l}{d_3}=20$, the area of the connecting pipe is $F_3=0.06F$, and the specific weight of the blast $\gamma_1=1.56$ kg. [0.0974 lbs. per cu. ft.]; then with a coefficient of friction of the air $\zeta=.024$ we obtain for the head due to this friction

$$\rho=\zeta\frac{l}{d_3}\frac{v_3^2}{2g}\gamma_1=\zeta\frac{l}{d_3}\left(\frac{F}{F_3}\right)^2\frac{v^2}{2g}\gamma_1=\frac{0.024\times20\times1.56}{0.06^2\times2\times9.81}v^2=10.6v^2\,[0.202v^2],$$

where we substitute for v^2 the mean square of the piston velocity v. Now the mean piston velocity is $\frac{15\times2\times1.6}{60}=0.8$ m. [2.62 ft.] and, according to Vol. II, the mean velocity square is $1.645\times0.8^2=1.053$ [11.33], so that $\rho=10.6\times1.053=11.16$ [2.29].

Hence the work consumed by the passage through the delivery-valve and the connecting pipe is

$$A_2=\left(\frac{G_2b_2}{F_2a_2}+\zeta\frac{l}{d_3}\frac{v_3^2}{2g}\gamma_1\right)Fs=(50+11.16)V=61.16V\,[12.52V\text{ ft.-lbs.}].$$

If we assume $\phi=\frac{1}{4}$ for the coefficient of friction for the piston-packing, and a breadth of packing $e=0.1$ m. [3.94 ins.], we get

$$4\phi\frac{e}{d}=\frac{0.1}{1.5}=0.067,$$

and finally the whole work needed for a single piston-stroke is

$$A=\left((1+4\phi\frac{e}{d}\right)(A_0+A_1+A_2)=1.067(2031.6+50+61.16)V$$
$$=2286V\text{ m.-kg. [}468V\text{ ft.-lbs. when }V\text{ is in cu. ft.];}$$

and since $V=2.827$ cu. m. [100 cu. ft.], we have

$$A=2286\times2.827=6462\text{ m.-kg. [46,740 ft.-lbs.].}$$

This value also gives the work per second, for the machine makes every minute $2\times2\times15=60$ single strokes, and therefore the number of horse-powers is given by $\frac{6462}{75}=86.2$.

The height z_1 which measured the excess of pressure of the outer air over that of the suction space is therefore

$$z_1 = \frac{G_1 b_1}{F_1 a_1 \gamma} = \frac{1}{2} \frac{100}{1.294} = 38.64 \text{ m. } [126.8 \text{ ft.}];$$

consequently, that this head may suffice to overcome the resistances of the air while passing through the suction-valves, the orifice of the suction-valve F_m' must at least be equal to

$$F_m' = \frac{F}{\alpha\left(\sqrt{\frac{G_1 b_1}{F_1 a_1 \gamma} \frac{2g}{v^2} - 1} + 1\right)} = \frac{F}{0.7\left(\sqrt{38.64 \frac{2 \times 9.81}{1.053} - 1} + 1\right)}$$
$$= \frac{F}{19.5} = 0.05F,$$

the coefficient of contraction α being taken equal to 0.7. The cross-section of suction-valves is usually much larger than the value given by this expression (see § 20).

§ 16. The Resistances of Blowers with Slide-valves.—Here the work theoretically necessary for the cross-section F of the cylinder and for each stroke of length s is also given by

$$\frac{x}{x-1}\left(\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}} - 1\right) pFs = A_0.$$

Hurtful resistances arise from the friction of the piston and of the slide-valve, from the passage of the air through the suction and delivery orifices, and from the friction of the air in the blast-pipe. The last-mentioned resistance is calculated as in blowers with clack-valves, but here the loss of pressure due to the passage of air through the valve-openings must be determined differently. For instance, in blowers with clack-valves a certain excess of pressure is needed to open the valves, while here the air-passages are opened by the crank-shaft. The resistances to the entrance and exit of the cylinder air are determined from the losses of head which, according to the laws of hydraulics, accompany a passage through contractions, and the magnitude of these resistances depends principally on the size of the cross-sections of the passages. Clack-valves open almost instantaneously, but slide-valves open the passages

gradually, consequently the losses of head vary during the piston-stroke; they can be determined in the following manner:

Let F_m be the area of the suction opening at any instant and α the coefficient of contraction; then the loss of head during suction is given by

$$\frac{(v_m-v_n)^2}{2g}=\frac{(v_m-v)^2}{2g}=\left(\frac{F}{\alpha F_m}-1\right)^2\frac{v^2}{2g}=\left(\frac{F}{\alpha bx}-1\right)^2\frac{v^2}{2g},$$

where b is the length of the suction-port and x the width of the port-opening given by the valve. Now, as αbx is always small in comparison with F, it will be accurate enough to replace the above expression by $\left(\frac{F}{\alpha bx}\right)^2\frac{v^2}{2g}$.

If we assume a small lap we can easily determine $\frac{v}{x}$ as follows: While the piston is traversing the distance $AM=\sigma$, Fig. 34, and the crank-pin is moving through the arc AO, the valve travels the distance $NQ=x$, and we have $\frac{x}{r_1}=\frac{y}{r}$, where r is the radius CA of the crank, r_1 the radius CB of the eccentric, and y the ordinate OM perpendicular to CA. Now, if v represents the velocity of the piston and c that of the crank-pin, we have $\frac{v}{c}=\frac{y}{r}$, and hence $\frac{v}{c}=\frac{x}{r_1}$ or $\frac{v}{x}=\frac{c}{r_1}$, so that the loss of head during suction is

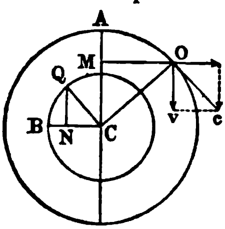

FIG. 34.

$$\left(\frac{F}{\alpha br_1}\right)^2\frac{c^2}{2g}.$$

With a small lap r_1 is nearly equal to the width a of the port; consequently the loss of head during suction is

$$\left(\frac{F}{\alpha F_1}\right)^2\frac{c^2}{2g},$$

where F_1 = area of port.

Therefore, if the atmospheric air after its passage through the suction-orifice is to follow the piston with a velocity v, there will be needed an excess of pressure z_1 of the atmospheric

pressure over the pressure p' of the suction-space in the cylinder, which is measured by the head

$$z_1=\left(\frac{F}{\alpha F_1}\right)^2\frac{c^2}{2g}+\frac{v^2}{2g},$$

so that the work needed for this purpose during a single stroke is

$$A_1=z_1\gamma Fs=\frac{\gamma Fs}{2g}\left(\left(\frac{F}{\alpha F_1}\right)^2c^2+v^2\right).$$

In like manner we may determine the loss of work done due to the passage of air out of the cylinder through the cavity in the valve to the delivery-pipe. If F_2 is the area of the current in the valve-space, and $v_2=\frac{F}{F_2}v$ the velocity of the air, the loss of head when the outlet is fully open is equal to

$$\left(\frac{F_2}{\alpha F_1}-1\right)^2\frac{v_2^2}{2g}=\left(\frac{F_2}{\alpha F_1}-1\right)^2\left(\frac{F}{F_2}\right)\frac{v^2}{2g}.$$

But here also there is a gradual opening and closing of the outlet; hence, as in the case of suction, we must express the piston velocity v in terms of the crank-pin velocity c and thus find the *average* loss of head.

$$\left(\frac{F_2}{\alpha F_1}-1\right)^2\left(\frac{F}{F_2}\right)^2\frac{c^2}{2g}=z_2'.$$

There is another loss of head due to the passage of the air from the valve-cavity, of cross-section F_2, to the blast-pipe, whose cross-section is F_3, which head is measured by

$$\left(\frac{F_3}{F_2}-1\right)^2\frac{v_3^2}{2g}=\left(\frac{F}{F_2}-1\right)^2\left(\frac{F}{F_3}\right)^2\frac{v^2}{2g}=z_2''.$$

Finally, the head needed to generate v_3 in the blast-pipe and to overcome its friction is measured by

$$\left(1+\zeta\frac{l}{d_3}\right)\frac{v_3^2}{2g}=\left(1+\zeta\frac{l}{d_3}\right)\left(\frac{F}{F_3}\right)^2\frac{v^2}{2g}=z_2'''.$$

If we let z_2 again represent the excess of pressure on the blowing side of the piston over that in the reservoir we can determine z_2 from the equation

$$z_2=z_2'+z_2''+z_2'''$$
$$=\left(\frac{F_2}{\alpha F_1}-1\right)^2\left(\frac{F}{F_2}\right)^2\frac{c^2}{2g}+\left(\frac{F_3}{F_2}-1\right)^2\left(\frac{F}{F_3}\right)^2\frac{v^2}{2g}+\left(1+\zeta\frac{l}{d_3}\right)\left(\frac{F}{F_3}\right)^2\frac{v^2}{2g}.$$

These resistances require during each stroke an amount of work $A_2=z_2\gamma_1Fs$, and, as in the preceding article, the total work for a single stroke of a double-acting blower with slide-valves is given by

$$A=\left(1+4\phi\frac{e}{d}\right)(A_0+A_1+A_2)$$
$$=\left(1+4\phi\frac{e}{d}\right)\left\{\frac{x}{x-1}\left(\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}-1\right)p+z_1\gamma+z_2\gamma_1\right\}Fs,$$

where z_1 and z_2 have the values determined above. The work needed to move the valve is not included in this, consequently it must be specially calculated.

Example.—Let us take the example given in the preceding article, but substitute slide-valves for clack-valves and assume $\frac{F_1}{F}=0.06$, $\frac{F_2}{F}=0.1$, $\frac{F_3}{F}=0.16$, $\frac{l}{d_3}=20$, and $\alpha=0.7$; then, since $v^2=1.645\times0.8^2$, $v^2=1.053$ [11.32], $c=\frac{\pi}{2}\times0.8=1.26$ [4.13 ft.], and $c^2=1.588$ [17.08], we get for z_1 and z_2 the following values:

$$z_1=\left(\frac{1}{0.7\times0.06}\right)^2 1.588\times0.051+1.053\times0.051=45.9+0.05$$
$$=46 \text{ m. [152 ft.]}.$$

$$z_2=\left(\frac{10}{0.7\times6}-1\right)^2 100\times1.588\times0.051$$
$$+(1.6-1)^2\left(\frac{100}{16}\right)^2 1.053\times0.051+(1+0.024\times20)\frac{1.053\times0.051}{0\ 16^2}$$
$$=15.44+0.75+3.01=19\ 2 \text{ m. [63 ft.]}.$$

Hence when $\gamma = 1.294$ kg. per cu. m. [0.08073 lbs. per cu. ft.] and $\gamma_1 = \frac{760+160}{760}\gamma = 1.56$ kg. per cu. m. [0.0974 lbs. per cu. ft.],

$$A_1 = z_1 \gamma F s = 46 \times 1.294 V = 59.5 V \ [12.2V],$$
$$A_2 = z_2 \gamma_1 F s = 19.2 \times 1.56 V = 29.95 V \ [6.13V];$$

and since $A_0 = 2031.6V$ [416V], $V = 2.827$ cu. m. [99.84 cu. ft.], and $1 + 4\phi\frac{e}{d} = 1.067$, then the whole work per stroke, which is here also equal to the work per second, is

$$A = 1.067(2031.6 + 59.5 + 29.95)2.827 = 6398 \text{ m.-kg.} \ [46,277 \text{ ft.-lbs.}],$$

and this corresponds to 85.3 horse-powers, almost exactly as in blowers with clack-valves.

§ 17. **Size of Blast-reservoirs.**—Another subject for calculation is the size of blast-reservoirs. In order to get the necessary basis let us assume a double-acting blower with one cylinder, and that the piston is driven by a uniformly rotating crank-pin, and that the blast flows uniformly from the reservoir, which last assumption is of course only approximately correct. The blower-piston K, Fig. 35, first traverses a certain distance $AE = s_1$, during which the air is compressed, but no air is forced into the reservoir. Let $s = 2r$ represent the whole piston-stroke AB, b the height of the barometer, and h the height of the manometer; then

$$s_1 = \frac{h}{b+h}2r; \quad \ldots \ldots \ldots \quad (1)$$

hence for the corresponding crank-angle $ACD = \theta$, at which the pressure on the cylinder becomes equal to that in the reservoir, we have

$$\cos\theta = \frac{r - s_1}{r} = 1 - \frac{2h}{b+h}, \quad \ldots \ldots \ldots \quad (2)$$

or

$$\sin\frac{\theta}{2} = \sqrt{\frac{h}{b+h}}. \quad \ldots \ldots \ldots \quad (3)$$

For another crank-angle $ACO=\beta$ and piston travel $AP=x=r(1-\cos\beta)$ the quantity of air introduced into the reservoir *per unit of piston area* is

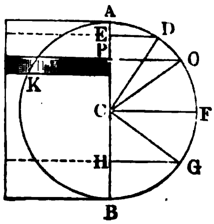

FIG. 35.

$$EP=x-s_1=r(1-\cos\beta)-s_1,$$

while the quantity which has left the reservoir, reduced to the density of the air in the latter, is

$$\frac{b}{b+h}\,\frac{AO}{AOB}2r=\frac{b}{b+h}\,\frac{\beta}{\pi}2r.$$

Consequently the *variable excess* of the volume that has left the reservoir over that which has entered it is

$$y=\frac{b}{b+h}\,\frac{\beta}{\pi}2r-r(1-\cos\beta)+s_1. \quad . \quad . \quad . \quad . \quad (4)$$

This is a maximum or minimum for $\frac{dy}{d\beta}=0$, i.e., for

$$\sin\beta=\frac{2}{\pi}\,\frac{b}{b+h},$$

or approximately

$$\sin\beta=\frac{2}{\pi}\left(1-\frac{h}{b}\right),$$

or for very small blast pressures

$$\sin\beta=\frac{2}{\pi}.$$

The acute angle (β_1) belonging to this sine corresponds to the maximum y_1, and the obtuse angle (β_2) to the minimum y_2 of y. When both these values are known we have for the greatest variation of blast volume in the reservoir the quantity y_1-y_2. If F is the piston area, and W the volume of the reservoir, then, according to *Mariotte's law*, the greatest variation z of the blast pressure is

$$\frac{b+h-z}{b+h}=\frac{W-F(y_1-y_2)}{W},$$

or

$$\frac{z}{b+h}=\frac{y_1-y_2}{W}F. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

If we designate the volume $F2r$ of the blowing-cylinder by V, we also have

$$\frac{z}{b+h}=\frac{y_1-y_2}{2Wr}V. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Now, if the ratio $\nu=\frac{z}{b+h}$ of the greatest difference of pressure to the average blast pressure is given, we have for the size of the reservoir

$$W=\frac{y_1-y_2}{2r}\frac{V}{\nu}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

or if the ratio $\delta=\frac{z}{h}$ of the difference z to the average height h of the manometer is given, then the necessary volume of the reservoir, taking equation (4) into account, is

$$W=\frac{y_1-y_2}{2r}\frac{b+h}{\delta h}V=\left(b\frac{\beta_1-\beta_2}{\pi}+(b+h)\times\frac{\cos\beta_1-\cos\beta_2}{2}\right)\frac{V}{\delta h}. \quad (9)$$

For very small pressures we have for $\sin\beta=\frac{\pi}{2}$ the angles

$$\beta_1=39° \, 32' \quad \text{and} \quad \beta_2=140° \, 28',$$

which give

$$y_1=0.2105r \quad \text{and} \quad y_2=-0.2105r,$$

and therefore determine

$$W=0.2105\frac{b}{\delta h}V. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

If we make $\theta=\beta_1$, we obtain, according to (3),

$$\frac{h}{b+h}=\left(\sin\frac{\beta_1}{2}\right)^2=(\sin 19° \, 46')^2=(0.3382)^2=0.1145.$$

Now, if θ is greater than β_1 or $\frac{h}{b+h}>0.1145$, then the maximum value of y occurs at the crank-angle θ; and since $r(1-\cos\theta)=s_1$, we have, according to (4), the maximum value

$$y_1=\frac{b}{b+h}\frac{\theta}{\pi}2r,$$

the minimum value y_2 remaining unchanged.

This determination also answers for two single-acting blowers, with cranks diametrically opposite, which force air into the same reservoir. On the other hand, if the blowing plant consists of two double-acting cylinders, a different formula must be employed, which may be found as follows:

Since one crank is here $\frac{\pi}{2}=90°$ in advance of the other, we must, according to (4), place

$$y=\frac{b}{b+h}\frac{\beta}{\pi}2r-r(1-\cos\beta)+s_1+\frac{b}{b+h}\left(\frac{\pi+2\beta}{\pi}\right)r-r(1+\sin\beta)+s_1$$

$$=\frac{b}{b+h}\left(1+\frac{4}{\pi}\beta\right)r-(2-\cos\beta+\sin\beta)r+2s_1, \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and from $\frac{dy}{d\beta}=0$ we obtain the maximum or minimum of this variable difference, for the equation reduces to

$$\sin\beta+\cos\beta=\frac{b}{b+h}\frac{4}{\pi},$$

or

$$\sin 2\beta=\left(\frac{b}{b+h}\frac{4}{\pi}\right)^2-1. \quad . \quad . \quad . \quad . \quad (12)$$

If we substitute the smaller value (β_1) belonging to this sign, in formula (11) we obtain the maximum value y_2; and if we introduce the larger angle (β_2), we obtain the minimum value; as for the work, the volume W is to be determined according to formula (9); hence we have

$$W=\left(\frac{2}{\pi}(\beta_1-\beta_2)b+[(\cos\beta_1-\cos\beta_2)-(\sin\beta_1-\sin\beta_2)]\frac{b+h}{2}\right)\frac{V}{\delta h}. \quad (13)$$

If $\beta_1<\theta$, we must determine y from θ and then

$$W=\left(\frac{2}{\pi}(\theta-\beta_2)b+[(\cos\theta-\cos\beta_2)-(\sin\theta-\sin\beta_2)]\frac{b+h}{2}\right)\frac{V}{\delta h}. \quad (13a)$$

For very small pressures we have $\sin 2\beta=\left(\frac{4}{\pi}\right)^2-1=0.621$; therefore

$$2\beta_1=38°\ 24' \quad \text{and} \quad 2\beta_2=141°\ 36',$$

or

$$\beta_1 = 19° \ 12' \quad \text{and} \quad \beta_2 = \ 70° \ 48',$$

and accordingly

$$y_1 = 0.0422r \quad \text{and} \quad y_2 = -0.0422r;$$

so that we now have

$$W = 0.0422\frac{b}{\delta h}V. \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Usually there is the *degree of regulation*

$$\delta = \frac{x}{h} = 0.04 \text{ to } 0.06.$$

Example.—Suppose a double-acting blowing-cylinder to furnish a blast of $h=50$ mm. [1.97 ins.] pressure from atmospheric air at 750 mm. [29.53 ins.] pressure; then the volume of the blast-reservoir, when the coefficient of fluctuation is $\delta=0.05$, must be

$$W = 0.2105\frac{b}{\delta h}V = 0.2105\frac{750}{0.05 \times 50}V = 63.15V \ [62.93V].$$

But when this blower produces a blast of 150 mm. [5.91 ins.] excess of pressure, that is, if

$$\frac{h}{b} = \frac{150}{750} = 0.2 \quad \text{and} \quad \frac{h}{b+h} = \frac{150}{900} = 0.1667,$$

the volume W must be determined according to formula (9) and $\beta_1 = \theta$ must be substituted in the formula. We then have

$$\sin\frac{\theta}{2} = \sqrt{\frac{h}{b+h}} = \sqrt{0.1667} = 0.4083;$$

hence

$$\frac{\theta}{2} = 24° \ 6' \quad \text{and} \quad \beta_1 = \theta = 48° \ 12';$$

on the other hand we have

$$\sin\beta_2 = \frac{2}{\pi}\frac{b}{b+h} = \frac{2}{\pi}\frac{750}{900} = 0.5305,$$

which gives

$$\beta_2 = 180° - 32° \, 4' = 147° \, 56'.$$

It now follows that

$$\frac{\beta_1 - \beta_2}{\pi} b = \frac{48.2 - 147.93}{180} \times 750 = -415.6 \text{ mm. } [-16.3 \text{ in.}],$$

and

$$\frac{\cos \beta_1 - \cos \beta_2}{2}(b+h) = 0.7568 \times 900 = 681.1 \text{ mm. } [26.81 \text{ ins.}];$$

hence the required volume of the blast-reservoir is

$$W = \left(\frac{\beta_1 - \beta_2}{\pi} b + \frac{\cos \beta_1 - \cos \beta_2}{2}(b+h) \right) \frac{V}{\delta h}$$

$$= (681.1 - 415.6) \frac{V}{0.05 \times 150} = 35.4V.$$

The reservoir does not regulate the blast perfectly, for piston friction requires a certain excess of force to set the piston in motion. If F is the area, G the load, and R the friction of the regulator-piston, the average excess of pressure of the blast in the regulator (reservoir) $p = \frac{G}{F}R$, the maximum value of the excess of pressure is

$$p_1 = \frac{G+R}{F},$$

and the maximum value

$$p_2 = \frac{G-R}{F};$$

consequently the greatest difference of blast pressure is

$$\delta = \frac{p_1 - p_2}{p} = \frac{2R}{G}.$$

Now $R = \phi \pi dep$, where d is the diameter of the regulator and e the width of its packing-ring; hence

$$\delta = \frac{2\phi \pi dep}{Fp} = 8\phi \frac{e}{d};$$

and if δ is given, as is usually the case, we have for the required *diameter of the regulator-piston*

$$d=8\phi\frac{e}{\delta}.$$

If we substitute

$$\phi=0.25,$$

then

$$d=2\frac{e}{\delta};$$

for example, for $\delta=0.05$

$$d=40e.$$

In the *bell-regulator*, with water packing, the resistances to motion are nearly 0 and the regulation is therefore a very perfect one.

But this is not the case with the so-called water-regulator, which consists of a stationary reservoir shut off below by water. Its efficiency is to be determined like that of the ordinary reservoir. As above, let $F(y_1-y_2)$ represent the greatest variation of the volume of the blast sent to the reservoir, and let the corresponding variation of the manometer be measured by a water column $=z$. Moreover, let the cross-section of the regulator proper, C, Fig. 36, be represented by G; let the cross-

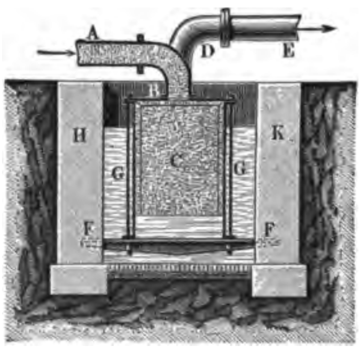

FIG. 36.

section of the reservoir HK containing C, after G has been subtracted, be represented by G_1; and let the rise or fall of the water-level in C corresponding to z be represented by x, also

the corresponding rise or fall of the water-level in HK by x_1; then

$$Gx=G_1x_1 \quad\text{and}\quad x_1=\frac{G}{G_1}x,$$

and

$$z=x+x_1=\frac{G+G_1}{G_1}x.$$

The increase in the blast-space due to the sinking of the water-level in C is

$$Gx=\frac{GG_1}{G+G_1}z.$$

We now have, as above,

$$\frac{b+h-z}{b+h}=\frac{W-[F(y_1-y_2)-Gx]}{W},$$

or

$$\frac{z}{b+h}=\frac{F(y_1-y_2)-Gx}{W},$$

and from this

$$\frac{z}{b+h}W=F(y_1-y_2)\frac{GG_1z}{G+G_1}.$$

If we substitute $z=\delta h$, we obtain

$$W=\left(F\frac{(y_1-y_2)}{\delta h}-\frac{GG_1}{G+G_1}\right)(b+h);$$

or if we represent the volume $2Fr$ of the blowing-cylinder by V, we get

$$W=\left(\frac{y_1-y_2}{2r}\frac{V}{\delta h}-\frac{GG_1}{G+G_1}\right)(b+h). \quad . \quad . \quad . \quad . \quad (15)$$

If a represents the average height of the cylinder-space C', then

$$W=Ga;$$

hence

$$a=\left(\frac{y_1-y_2}{2r}\frac{V}{\delta Gh}-\frac{G_1}{G+G_1}\right)(b+h). \quad . \quad . \quad . \quad . \quad (16)$$

Here b and h, as well as z, x, and x_1, are expressed in water columns (heads of water).

In order to obtain as small a reservoir as possible $\frac{G_1}{G}$ must be very large, i.e., the blast-chamber must be located in a large body of water, say a pond, then $\frac{G_1}{G+G_1}=1$, and therefore

$$W=\left(\frac{y_1-y_2}{2r}\frac{V}{\delta h}-G\right)(b+h),$$

or

$$W\left(1+\frac{b+h}{a}\right)=\frac{y_1-y_2}{2r}\frac{b+h}{\delta h}V;$$

hence

$$W=\frac{y_1-y_2}{2r}\frac{b+h}{\delta h}\frac{V}{1+\frac{b+h}{a}}. \quad . \quad . \quad . \quad . \quad (17)$$

Example.—In the dry reservoir above calculated we found for $b=750$ mm. [29.53 ins.], $h=150$ mm. [5.91 ins.], and $\delta=0.05$, the volume

$$\frac{y_1-y_2}{2r}\frac{b+h}{\delta}V=35.4V.$$

But if we use in its place a water-regulator and locate it in a large body of water, the average blast-space of the regulator having a height $a=2$ m. [6.56 ft.], then the volume of this space need only be

$$W=\frac{35.4V}{1+\frac{0.90\times 13.6}{2}}=\frac{35.4}{7.12}V=4.97V.$$

§ 18. **Size of the Tuyeres.**—From the volume of the blast delivered by the blower to the blast-reservoir (regulator), we can determine the necessary cross-section of the tuyere-orifice (F_u). If the tuyeres are directly attached to the blast-reservoir, the volume discharged, measured under the outer pressure p, will be given by a formula developed in Vol. I, which is

$$Q=uF_u\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}\sqrt{2g\frac{p_1}{\gamma_1}\frac{x}{x-1}\left(1-\left(\frac{p}{p_1}\right)^{\frac{x-1}{x}}\right)}$$

$$=uF_u\left(\frac{b+h}{b}\right)^{0.296}\sqrt{2g\frac{p_1}{\gamma_1}\times 3.38\left(1-\left(\frac{b}{b+h}\right)^{0.296}\right)}, \quad . \quad (1)$$

and for the metric units [English units] we substitute

$$\sqrt{2g\frac{p_1}{\gamma_1}}=396\sqrt{1+0.00367t}\quad[1300\sqrt{1+0.00203(t-32)}],$$

where t is the temperature of the blast in the reservoir. This temperature, if the blast is not heated, can be placed equal to that of the outer air, for, although the air is heated by compression from p to p_1, it loses this heat by the cooling action of the walls of the blowing-cylinder, blast-pipe, and reservoir.

If t_1 is the heat of the air after compression and t the temperature of the air sucked in, then

$$\frac{1+\delta t_1}{1+\delta t}=\left(\frac{b+h}{b}\right)^{\frac{x-1}{x}}=\left(1+\frac{h}{b}\right)^{0.296}.$$

Approximately

$$\left(\frac{b+h}{b}\right)^{\frac{x-1}{x}}=1+\frac{x-1}{x}\frac{h}{b}-\frac{x-1}{2x^2}\left(\frac{h}{b}\right)^2;$$

hence

$$t_1=t+\frac{x-1}{x}\frac{h}{b}\left(1-\frac{1}{2x}\frac{h}{b}\right)\left(t+\frac{1}{\delta}\right);$$

or, since

$$\delta=0.00367\ [0.00203]\ \text{and}\ x=1.42,$$

we have

$$t_1=t+0.296\frac{h}{b}\left(1-0.352\frac{h}{b}\right)(t+273^\circ)$$

$$\left[t_1=t+0.296\frac{h}{b}\left(1-0.352\frac{h}{b}\right)(t+459.4)\right];$$

for example, for $\frac{h}{b}=0.1$,

$$t_1=t+0.0285(t+273^\circ)\quad[t_1=t+0.0285(t+459.4)];$$

hence the increase of temperature is

$$t_1-t=0.0285(t+273^\circ)=7.78^\circ+0.0285t\ [13.1^\circ+0.0285t];$$

consequently for ordinary temperatures t_1-t is about 8° C. [14° F.]. However, to make sure of a sufficient margin it is well to assume the temperature of the blast in the reservoir only $\frac{t_1-t}{2}$ degrees greater than that of the atmosphere, that is, for $\frac{h}{b}=0.1$ about 4° C. [7° F.] higher than the outer air.

For ordinary blast pressures it is more convenient to transform the above expression for Q into the following approximate formula. When $\frac{x-1}{x}=n$ we have

$$\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}\sqrt{\frac{x}{x-1}\left(1-\left(\frac{p}{p_1}\right)^{\frac{x-1}{x}}\right)}=\sqrt{\frac{1}{n}\left(\frac{p_1}{p}\right)^n\left(\left(\frac{p_1}{p}\right)^n-1\right)}$$

$$=\sqrt{\frac{1}{n}\left(1+\frac{h}{b}\right)^n\left(\left(1-\frac{h}{b}\right)^n-1\right)}$$

$$=\sqrt{\left(1+n\frac{h}{b}+n\frac{n-1}{2}\left(\frac{h}{b}\right)^2\right)\left(1+\frac{n-1}{2}\frac{h}{b}+\frac{n-1}{2}\frac{n-2}{3}\left(\frac{h}{b}\right)^2\right)\frac{h}{b}}$$

$$=\sqrt{\left(1+\frac{3n-1}{2}\frac{h}{b}+\frac{7n^2-9n+2}{6}\left(\frac{h}{b}\right)^2\right)\frac{h}{b}};$$

or if we substitute $\frac{x-1}{x}$ for n again, we get

$$\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}\sqrt{\frac{x}{x-1}\left(1-\left(\frac{p}{p_1}\right)^{\frac{x-1}{x}}\right)}=\left(1+\frac{2x-3}{2x}\frac{h}{b}-\frac{5x-7}{6x^2}\left(\frac{h}{b}\right)^2\right)\frac{h}{b};$$

hence the volume of blast blowing out is

$$Q=uF_u\sqrt{2g\frac{p_1}{\gamma_1}\frac{h}{b}\left(1+\frac{2x-3}{2x}\frac{h}{b}-\frac{5x-7}{6x^2}\left(\frac{h}{b}\right)^2\right)}, \quad . \quad (2)$$

If we introduce $x=1.42$, we obtain

$$Q=uF_u\sqrt{2g\frac{p_1}{\gamma_1}\frac{h}{b}\left(1-0.0563\frac{h}{b}-0.0083\left(\frac{h}{b}\right)^2\right)}.$$

For small blast pressures we may use the simpler formula

$$Q = uF_u\sqrt{2g\frac{p_1}{\gamma_1}\frac{h}{b}\left(1-0.0563\frac{h}{b}\right)}$$
$$= uF_u\left(1-0.028\frac{h}{b}\right)\sqrt{2g\frac{p_1}{\gamma_1}\frac{h}{b}}, \quad . \quad . \quad . \quad . \quad (2a)$$

or the still simpler formula

$$Q = uF_u\sqrt{2g\frac{p_1}{\gamma_1}\frac{h}{b}} = uF_u\sqrt{2g\frac{p}{\gamma}\frac{h}{b}}; \quad . \quad . \quad . \quad (2b)$$

or finally, if the density of the manometer fluid is e times as great as the density of the outer air, so that we may place $p = eb\gamma$, we have

$$Q = uF_u\sqrt{2geh}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (2c)$$

exactly as with an incompressible fluid (see Vol. I).

We can now determine from the blast volume Q *the cross-section of the tuyere-orifice*, or, if there are several tuyeres, the sum of the cross-sections of all their orifices, according to the formula

$$F_u = \frac{Q}{u\left(1-0.028\frac{h}{b}\right)\sqrt{2g\frac{p}{\gamma}\frac{h}{b}}} = \frac{\left(1+0.028\frac{h}{b}\right)Q}{u\sqrt{2g\frac{p}{\gamma}\frac{h}{b}}};$$

i.e., for metric units

$$F_u = \frac{\left(1+0.028\frac{h}{b}\right)Q}{396u\sqrt{(1+0.00367t)\frac{h}{b}}} \text{ sq. m.}, \quad . \quad . \quad . \quad (3)$$

and for English units

$$F_u = \frac{\left(1+0.028\frac{h}{b}\right)Q}{1300u\sqrt{1+0.00203(t-32)\frac{h}{b}}} \text{ sq. ft.}$$

We may assume that the actual volume of blast furnished by a cylinder-blower is only 60 to 75 per cent. of the theoretical quantity $\frac{nFs}{60}$, and accordingly we must substitute in the above

formulas $0.60Q$ to $0.75Q$ in place of Q. The coefficient of efflux u is not perfectly constant, being 0.91 for small manometer heights (h) of 1 cm. [0.39 in.] and 0.928 for great manometer heights (h) of 20 cm. [7.87 ins.], but on an average we may assume $u=0.92$. From the cross-section F_u just found and the number of tuyeres we can easily determine the diameters of the tuyere-orifices.

Example.—If the blower calculated in § 15 offers 2.827 cu. m. [99.84 cu. ft.] space per second to the air, we may assume that the actual volume of the blast, measured under atmospheric pressure, per second is

$$Q=0.7\times 2.827=1.98=2 \text{ cu. m. [70 cu. ft.]}.$$

If the height of the barometer is $b=760$ mm. [29.92 ins.], the blast pressure $h=160$ mm. [6.3 ins.], the temperature of the blast $t=15°$ C. [59° F.], and the coefficient of efflux $u=0.92$, then the necessary cross-section of the tuyeres is

$$F_u=\frac{\left(1+0.028\frac{160}{760}\right)2}{396\times 0.92\sqrt{(1+0.00367\times 15)\frac{160}{760}}}=\frac{2.0118}{364.3\sqrt{0.2221}}$$
$$=0.0117 \text{ sq. m. [0.126 sq. ft.]}.$$

If there are three tuyeres and their orifices are circular, we have for the diameter of one orifice

$$d_u=\sqrt{\frac{4F_u}{3\pi}}=0.070 \text{ m. [2.76 ins.]}.$$

In the method of determination above adopted it was assumed that the height h of the manometer was given for the blast in the reservoir; but if the manometer is placed at the end of the blast-pipe close to the tuyere, it has a lower value h_1, because a part of the original pressure has been expended in generating the velocity of the air in the blast-pipe.

Let F_1, v_1, and γ_1 be respectively the cross-section of pipe, the velocity and the density of the blast at the point where the manometer is placed, and F_u, v_u, and γ_u the cross-section of the tuyere, the velocity and the density of the blast at the outlet; then the weight of the blast discharged is

$$F_1 v_1 \gamma_1 = F_u v_u \gamma_u,$$

and therefore

$$\frac{v_1}{v_u}=\frac{F_u}{F_1}\frac{\gamma_u}{\gamma_1}=\frac{F_u}{F_1}\left(\frac{b}{b+h_1}\right)^{\frac{1}{x}};$$

hence the head needed to increase the velocity from v_1 to v_u is

$$\frac{v_u^2-v_1^2}{2g}=\left(1-\left(\frac{F_u}{F_1}\right)^2\left(\frac{b}{b+h_1}\right)^{\frac{2}{x}}\right)\frac{v_u^2}{2g}=C\frac{v_u^2}{2g}.$$

This value must be placed equal to $\frac{p_1}{\gamma_1}\frac{x}{x-1}\left(1-\left(\frac{b}{b+h_1}\right)^{\frac{x-1}{x}}\right)$ (see Vol. I), and then we get the corresponding *velocity of efflux*,

$$v_u=\frac{1}{\sqrt{C}}\sqrt{2g\frac{p_1}{\gamma_1}\frac{x}{x-1}\left(1-\left(\frac{b}{b+h_1}\right)^{\frac{x-1}{x}}\right)},$$

and the *discharge*, measured under atmospheric pressure, is

$$Q=\frac{uF_u}{\sqrt{C}}\left(\frac{p_1}{p}\right)^{\frac{x-1}{x}}\sqrt{2g\frac{p_1}{\gamma_1}\frac{x}{x-1}\left(1-\left(\frac{b}{b+h_1}\right)^{\frac{x-1}{x}}\right)}. \quad . \quad (4)$$

We may use the approximation found above and obtain the more convenient formula

$$Q=uF_u\frac{1-0.028\frac{h_1}{b}}{\sqrt{C}}\sqrt{2g\frac{p_1}{\gamma_1}\frac{h_1}{b}}, \quad . \quad . \quad . \quad . \quad (4a)$$

or, if we further approximate by placing $C=1-\left(\frac{F_u}{F_1}\right)^2$, we get the simpler, and for most cases sufficiently accurate, formula

$$Q=\left(1-0.028\frac{h_1}{b}\right)uF_u\sqrt{\frac{2g\frac{p_1}{\gamma_1}\frac{h_1}{b}}{1-\left(\frac{F_u}{F_1}\right)^2}}. \quad . \quad . \quad . \quad (4b)$$

If in this we substitute

$$\sqrt{2g\frac{p_1}{\gamma_1}}=396\sqrt{1+\delta t}\left[=1300\sqrt{1+0.00203(t-32)}\right],$$

we obtain the required cross-section of the tuyere

$$F_u=\left(1+0.028\frac{h_1}{b}\right)\frac{Q}{396u}\sqrt{\frac{1-\left(\frac{F_u}{F_1}\right)^2}{(1+0\ 00367t)\frac{h_1}{b}}}\text{ sq. m.} \quad . \quad (5)$$

$$\left(F_u=\left(1+0.028\frac{h_1}{b}\right)\frac{Q}{1300u}\sqrt{\frac{1-\left(\frac{F_u}{F_1}\right)^2}{[1+0.00203(\ -32)]\frac{h_1}{b}}}\text{ sq. ft.}\right).$$

If the reservoir is connected with the tuyere by a *long pipe*, the friction of the blast in the pipe must be taken into account.

Let l_1 be the length, d_1 the diameter, and F_1 the cross-section of this pipe; the head which measures the friction of the blast in the pipe is

$$z=\zeta_1\frac{l_1}{d_1}\left(\frac{F_u}{F_1}\right)^2\frac{v_u^2}{2g}=\zeta_1\frac{l_1}{d_1}\left(\frac{d_u}{d_1}\right)^4\frac{v_u^2}{2g},$$

where the coefficient of resistance ζ_1 may be taken equal to 0.025.

Besides this resistance there are others in the blast-pipe, namely, resistance at bends, at cocks, etc. These may be taken into account as in water-pipes, the lost pressure due to suddenly transforming the velocity v_1 into v_2 being here also

$$z_1=\frac{(v_1-v_2)^2}{2g}, \text{ etc.}$$

If we represent the coefficient of resistance by ζ_u and the sum of all the coefficients of the remaining resistances to the motion of the air in the blast-pipe by $\Sigma(\zeta)$, we get,

$$Q=\left(1-0.028\frac{h}{b}\right)uF_u\sqrt{\frac{2g\frac{p_1}{\gamma_1}\frac{h}{b}}{1+\zeta_u+\left(\Sigma(\zeta)+\zeta_1\frac{l_1}{d_1}\right)\left(\frac{F_u}{F_1}\right)^2}}$$

$$=396\left(1-0.028\frac{h}{b}\right)uF_u\sqrt{\frac{(1+0.00367t)\frac{h}{b}}{1+\zeta_u+\left(\Sigma(\zeta)+\zeta_1\frac{l_1}{d_1}\right)\left(\frac{F_u}{F_1}\right)^2}}, \quad (6)$$

and therefore the cross-section of the tuyere is

$$F_u = \left(1+0.028\frac{h}{b}\right)\frac{Q}{396u}\sqrt{\frac{1+\zeta_u+\left(\Sigma(\zeta)+\zeta_1\frac{l_1}{d_1}\right)\left(\frac{F_u}{F_1}\right)^2}{(1+0.00367t)\frac{h}{b}}}\text{ sq. m.} \quad (7)$$

$$\left[F_u = \left(1+0.028\frac{h}{b}\right)\frac{Q}{1300u}\sqrt{\frac{1+\zeta_u+\left(\Sigma(\zeta)+\zeta_1\frac{l_1}{d_1}\right)\left(\frac{F_u}{F_1}\right)^2}{\left(1+0.00203(t-32)\right)\frac{h}{b}}}\text{ sq. ft.}\right].$$

If a blast-pipe separates into branches as, for example, with the furnace shown in Fig. 37, where the blast is admitted to

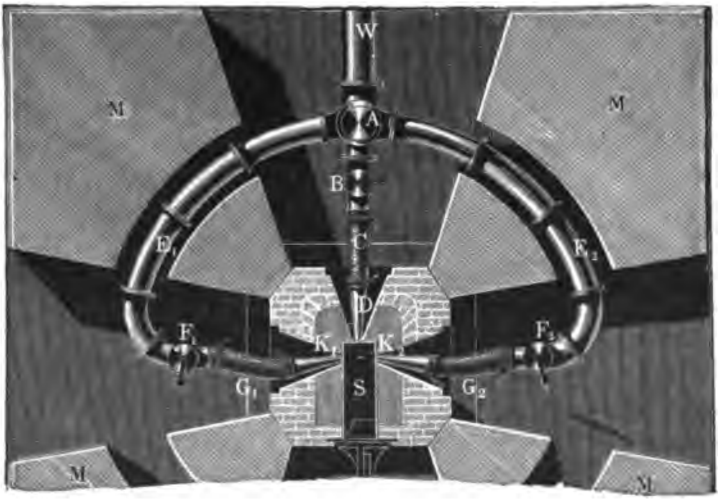

FIG. 37.

the smelting-space through three tuyeres, then the calculation of F_u must be determined as in the branch-pipes of water-pipes (see Vol. II).

If z is the (usually unknown) height of the manometer just in front of where the branching takes place, l_1, d_1, and F_1 the length, diameter, and cross-section of the main pipe, measured from the reservoir to the branching-off point, ζ_0 the coeffi-

cient of resistance at the entrance, and v_1 the velocity of the blast in the pipe, we have the formula

$$z=h-\left(1+\zeta_0+\zeta_1\frac{l_1}{d_1}\right)\frac{v_1^2}{2g}=h-\frac{1}{2ge}\left(1+\zeta_0+\zeta_1\frac{l_1}{d_1}\right)\left(\frac{4Q}{\pi d_1^2}\right)^2, \quad (8)$$

and by substituting the value of z for h in formula (7) we get

$$F_u=\left(1+0.028\frac{z}{b}\right)\frac{Q}{396nu}\sqrt{\frac{1-\left(\frac{F_u}{F_2}\right)^2+\zeta_u+\left(\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)\left(\frac{F_u}{F_2}\right)^2}{1+0.00367t\frac{z}{b}}}$$

$$\left[F_u=\left(1+0.028\frac{z}{b}\right)\frac{Q}{1300nu}\sqrt{\frac{1-\left(\frac{F_u}{F_2}\right)^2+\zeta_u+\left(\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)\left(\frac{F_u}{F_2}\right)^2}{[1+0.00203(t-32^\circ)]\frac{z}{b}}}\right]$$

where l_2, d_2, etc., represent the length, diameter, etc., of one of the n branch-pipes and F_u the cross-section of the tuyere-orifice.

Example.—Suppose that in the blast arrangement shown in Fig. 37 the blast in the main pipe WA, 5 m. [16 ft.] long, moves with a velocity $v_1=10$ m. [32.8 ft.], and that the volume of blast, as in the preceding example, is $Q=2$ cu. m. [70.6 cu. ft.]; then the cross-section of these pipes will be

$$F_1=\frac{Q}{v_1}=0.2 \text{ sq. m. [2.15 sq. ft.]},$$

which corresponds to a diameter of

$$d_1=\sqrt{\frac{4\times0.2}{3.14}}=0.505 \text{ m.}=0.5 \text{ m. [19.88 in.]}.$$

If we assume the coefficient of resistance for the entrance to the pipe to be $\zeta_0=0.5$, the coefficient of friction $\zeta=0.025$, the height of the manometer in the reservoir $h=160$ mm. [6.3 ins.], and the specific weight of the blast, relatively to mercury, to be

$$\frac{1}{e}=\frac{1}{800\times13.6}=\frac{1}{10,880},$$

then the pressure at the branching-off point or end A of this pipe is found from formula (8) to be

$$z=h-\left(1+\zeta_0+\zeta_1\frac{l_1}{d_1}\right)\frac{v_1^2}{2ge}=0.160-\left(1.5+0.025\frac{5}{0.5}\right)\frac{10^2}{10.88}0.051$$
$$=(0.160-0.0008)\text{ m.}=159.2\text{ mm. [6.27 ins.]}.$$

According to the preceding example, the area of the three tuyere-orifices is approximately $3F_u=0.0117$ sq. m. [0.126 sq. ft.]; consequently the cross-section of one orifice is $F_u=0.0039$ sq. m. [0.042 sq. ft.], and its diameter 0.07 m. [2.76 ins.]. If we make the cross-section of a branch-pipe

$$F_2=\frac{F_1}{3}=\frac{0.2}{3}=0.0667\text{ sq. m. [0.718 sq. ft.]},$$

and accordingly make the diameter $d_2=0.292$ m. [11.5 ins.], we have

$$1-\left(\frac{F_u}{F_2}\right)^2=1-\left(\frac{0.0039}{0.0667}\right)^2=1-0.0034=0.9966;$$

and if we assume $\Sigma(\zeta)=2$, also $\zeta_2=0.025$, and the length of a branch-pipe $l_2=10$ m. [32.8 ft.], we get

$$\left(\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)\left(\frac{F_u}{F_2}\right)^2=\left(2+0.025\frac{10}{0.292}\right)0.0034=0.01;$$

and as

$$\zeta_u=\left(\frac{1}{0.92}\right)^2-1=0.1814,$$

$$\left(1+0.028\frac{z}{b}\right)=1+0.028\frac{159.2}{760}=1.006;$$

also, for $t=15°$ C. [59° F.],

$$(1+0.00367t)\frac{z}{b}=1.055\frac{159.2}{760}=0.221,$$

and

$$\frac{Q}{396nu}=\frac{2}{396\times3\times0.92}=0.00183\ [0.0197];$$

then we have, according to formula (9), for the necessary cross-section of a tuyere-orifice

$$F_u = 1.006 \times 0.00183 \sqrt{\frac{0.9966 + 0.1814 + 0.010}{0.221}}$$
$$= 0.00427 \text{ sq. m. } [0.046 \text{ sq. ft.}],$$

which corresponds to a diameter of

$$d_u = \sqrt{\frac{4 \times 0.00427}{3.14}} = 0.074 \text{ m. } [2.91 \text{ ins.}].$$

§ 19. The Hot-air Blast.—When a system of pipes is employed to heat the blast, the latter must also overcome the friction in these pipes and the resistances at the bends. These resistances are not insignificant, for, in order that the heat may be quickly imparted, the separate pipes have but a small hydraulic mean depth $\left(d = \frac{4F}{p}\right)$. In such a system of pipes the temperature t_1 of the air coming from the reservoir is gradually changed to the higher temperature t_2, from 200° to 300° C. [390° to 570° F.], its density changing from γ_1 to a smaller value γ_2, and the velocity v_1 becoming v_2. As the weight of the blast flowing through the cross-section F of the pipes is represented by $Fv_1\gamma_1 = Fv_2\gamma_2$, we have

$$\frac{v_2}{v_1} = \frac{\gamma_1}{\gamma_2}.$$

If the height of the manometer at the end of the heater-pipes is smaller by the amount y than the height h at the beginning, then, according to Vol. I, we have

$$\frac{\gamma_1}{\gamma_2} = \frac{b+h}{b+h-y} \frac{1+\delta t_2}{1+\delta t_1},$$

hence we also have approximately

$$\frac{v_2}{v_1} = \left(1 + \frac{y}{b+h}\right) \frac{1+\delta t_2}{1+\delta t_1}; \quad . \quad . \quad . \quad . \quad . \quad (1)$$

so that, if we abbreviate by making

$$\frac{1+\delta t_2}{1+\delta t_1}=x, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

we get

$$\left(\frac{v_2}{v_1}\right)^2=\left(1+\frac{2y}{b+h}\right)x^2,$$

and

$$v_2^2-v_1^2=\left(x^2-1+\frac{2y}{b+h}x^2\right)v_1^2, \quad . \quad . \quad . \quad . \quad (3)$$

also

$$v_2^2+v_1^2=\left(x^2+1+\frac{2y}{b+h}x^2\right)v_1^2. \quad . \quad . \quad . \quad . \quad (4)$$

If $\Sigma(\zeta)$ represents the sum of the coefficients of the resistances at the bends, ζ_2 the coefficient of friction, l_2 the length, and d_2 the diameter of the heater-pipes, the head needed to overcome the resistances in these pipes is

$$\begin{aligned} q&=\left(\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)\frac{\frac{1}{2}(v_1^2+v_2^2)}{2g}\\ &=\left(\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)\left(\frac{x^2+1}{2}+\frac{y}{b+h}x^2\right)\frac{v_1^2}{2g}; \quad . \quad . \quad . \quad (5)\end{aligned}$$

and if ε is the ratio of the density of mercury to that of the outer air at pressure b and temperature t_1, then

$$\begin{aligned} \varepsilon y&=\frac{v_2^2-v_1^2}{2g}+q\\ &=\left(x^2-1+\frac{2y}{b+h}x^2\right)\frac{v_1^2}{2g}+\left(\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)\left(\frac{x^2+1}{2}+\frac{y}{b+h}x^2\right)\frac{v_1^2}{2g},\end{aligned}$$

or

$$\left\{\varepsilon-\left(2x^2+\left(\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)x^2\right)\frac{v_1^2}{2g(b+h)}\right\}y$$
$$=\left(x^2-1+\left(\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)\frac{x^2+1}{2}\right)\frac{v_1^2}{2g},$$

so that the lowering of the height of the manometer on the heater-pipes is

$$y=\frac{x^2-1+\left(\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)\frac{x^2+1}{2}}{\varepsilon-\left(2+\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}\right)x^2\frac{v_1^2}{2g(b+h)}}\frac{v_1^2}{2g}; \quad . \quad . \quad (6)$$

and now the velocity of the hot blast at the end of the heater-pipes can be determined by means of formula (1):

$$v_2=\left(1+\frac{y}{b+h}\right)xv_1=\left(1+\frac{y}{b+h}\right)\left(\frac{1+\delta t_2}{1+\delta t_1}\right)v_1. \quad . \quad . \quad (7)$$

If, as above, h represents the height of the manometer on the reservoir, d_1 the diameter, and l_1 the length of the blast-pipe conducting the air from the reservoir to the heater, etc., the height of the manometer at the end of the heating apparatus is

$$z=h-y-\left(1+\zeta_0+\zeta_1\frac{l_1}{d_1}\right)\frac{v_1^2}{2g\varepsilon}. \quad . \quad . \quad . \quad . \quad (8)$$

The volume of air discharged through the tuyeres with a temperature t_2 and reduced to the outer pressure b is

$$Q_1=\left(1\pm 0.028\frac{z}{b}\right)uF_u\sqrt{\frac{2g\frac{p_1}{\gamma_1}\frac{z}{b}}{1-\left(\frac{F_u}{F_3}\right)^2+\zeta_u+\left(\Sigma(\zeta)+\zeta_3\frac{l_3}{d_3}\right)\left(\frac{F_u}{F_3}\right)^2}}, \quad (9)$$

where l_3, d_3, and F_3 are respectively the length, diameter, and cross-section of the branch-pipes leading to the tuyeres. Here ζ_3 represents the coefficient of friction, $\Sigma(\zeta_1)$ the sum of the coefficients of all the rest of the resistances of the branch pipes, and $\zeta_u=\frac{1}{u^2}-1$ the coefficient of resistance of the tuyere.

If we abbreviate by placing

$$1-\left(\frac{F_u}{F_3}\right)^2+\zeta_u+\left(\Sigma(\zeta_1)+\zeta_3\frac{l_3}{d_3}\right)\left(\frac{F_u}{F_3}\right)^2=w, \quad . \quad . \quad (10)$$

and

$$\sqrt{2g\frac{p_1}{\gamma_1}}=396\sqrt{1+\delta t_2}\ [=1300\sqrt{1+\delta t_2}],$$

we can then write

$$Q_1=396\left(1-0.028\frac{z}{b}\right)uF_u\sqrt{\frac{1+\delta t_2}{w}\frac{z}{b}}.$$

Therefore the volume of the blast reduced to the outer temperature t and the barometric pressure b becomes

$$Q=\frac{1+\delta t}{1+\delta t_2}Q_1=396(1+\delta t)\left(1-0.028\frac{z}{b}\right)uF_u\sqrt{\frac{z}{(1+\delta t_2)wb}}. \quad (11)$$

$$\left[Q=1300(1+\delta t)\left(1-0.028\frac{z}{b}\right)uF_u\sqrt{\frac{z}{(1+\delta t_2)wb}}\right],$$

so that for a given volume of blast the total area of the orifices of all the tuyeres is

$$F_u=\frac{1+0.028\frac{z}{b}}{396(1+\delta t)u}Q\sqrt{(1+\delta t_2)w\frac{b}{z}} \text{ sq. m.} \quad . \quad . \quad (12)$$

$$\left[Fu=\frac{1+0.028\frac{z}{b}}{1300(1+\delta t)u}Q\sqrt{(1+\delta t_2)w\frac{b}{z}} \text{ sq. ft.}\right].$$

Consequently the orifices of the tuyeres must have a greater cross-section the lower the temperature t of the outer air and the higher the temperature t_2 of the hot blast.

In order to institute comparisons as to the oxygen forced into the furnace by the blower, the volume of the blast is reduced to 0° C. [32° F.], and to the average barometric pressure $b_0=760$ m.m. [29.92 ins.]. This reduced volume of the blast is

$$Q_0=\frac{1}{1+\delta t_2}\frac{b}{b_0}Q_1=\frac{1}{1+\delta t}\frac{b}{b_0}Q. \quad . \quad . \quad . \quad . \quad . \quad (13)$$

In order that the volume Q of the blast may be heated from the temperature t_1 to the desired temperature t_2 while flowing through the heated pipes, it is necessary to give the latter a certain heating surface O. According to *Cavé* 1 sq. m. [10.76 sq. ft.] of heating surface will evaporate 19 kg. of water [42 lbs.] per hour, that is, transmit $\frac{19\times 600}{60}=190$ calories [754 B.T.U.] per minute. [This is equivalent to 70 units of heat per sq. ft.] According to *Walter*, estimates of the heating surface of hot-

blast stoves should be based on a transmission of 100 calories per square meter [about 37 B.T.U. per sq. ft.] per minute. As the specific heat of air is only a quarter that of water, 1 sq. m. of heating surface will heat 1 kg. of air to about 400° C., or $\frac{3}{4}$ kg., i.e., 1 cu. m. to about 300° C. per minute; consequently *1 sq. m. of heating surface per minute will heat about 1 cu. m. of air to 300° C.* [This is nearly equivalent to 1 sq. ft. of heating surface raising 1 cu. ft. of air per minute to a temperature of 1000° F.]

Let p_w represent the outer circumference and l_w the length of the heating-pipes; as the hot air comes in contact with the outside of the pipes we must make, according to the above,

$$p_w l_w = 60Q \text{ meters (when air is heated to 300° C.),}$$

where Q is the quantity of air to be heated per second; a more general expression is

$$p_w l_w = 0.2(t_2 - t_1)Q\,[0.034(t_2 - t_1)Q];$$

accordingly the total length of all the heating-pipes is

$$l_w = 0.2(t_2 - t_1)\frac{Q}{p_w} \text{ meters} \left[= 0.034(t_2 - t_1)\frac{Q}{p_w} \right].$$

If a heating-pipe has an *elliptical* cross-section with the semi-axes a_2 and $b_2 = \nu a_2$, the inner cross-section is

$$F_2 = \pi a_2 b_2 = \nu \pi a_2^2;$$

hence

$$a_2 = \sqrt{\frac{F_2}{\nu\pi}}, \quad \text{also} \quad b_2 = \nu\sqrt{\frac{F_2}{\nu\pi}} = \sqrt{\frac{\nu F_2}{\pi}}.$$

Moreover, its inner circumference is approximately

$$p_2 = \pi(a_2 + b_2)\left(1 + \frac{1}{4}\left(\frac{a_2 - b_2}{a_2 + b_2}\right)^2\right);$$

hence

$$\frac{p_2}{F_2} = \frac{a_2 + b_2}{a_2 b_2}\left(1 + \frac{1}{4}\left(\frac{a_2 - b_2}{a_2 + b_2}\right)^2\right).$$

If the pipes have a thickness e_2, their outer perimeter is

$$p_w=\pi(a_2+b_2+2e_2)\left(1+\frac{1}{4}\left(\frac{a_2-b_2}{a_2+b_2+2e_2}\right)^2\right).$$

Example.—Let us suppose a hot-blast stove of the kind shown in Fig. 24, and containing six series of pipes to be inserted in the blast arrangements discussed in the example given in § 18. Let the inner height of the elliptical section of such a pipe be six times its width; then, as the area of the blast-pipe is 0.2 sq. m. [2.15 sq. ft.], the cross-section of each pipe is $F_2=0.0333$ sq. m. [0.362 sq. ft.]. Accordingly, the greatest inside diameter is

$$2a_2=2\sqrt{\frac{F_2}{\pi\nu}}=2\sqrt{\frac{0.0333\times 6}{3.14}}=0.504 \text{ m. } [1.65 \text{ ft.}],$$

and the least is

$$2b_2=\frac{0.504}{6}=0.084 \text{ m. } [0.276 \text{ ft.}];$$

consequently the inner periphery is

$$p_2=\pi(a_2+b_2)\left(1+\frac{1}{4}\left(\frac{a_2-b_2}{a_2+b_2}\right)^2\right)$$

$$=3.14\times 0.294\left(1+\frac{1}{4}\left(\frac{0.210}{0.294}\right)^2\right)=1.041 \text{ m. } [3.41 \text{ ft.}];$$

hence the ratio

$$\frac{p_2}{F_2}=\frac{1.041}{0.0333}=31.23.$$

If the pipes have a thickness of 15 mm. [0.59 in.],

$$a_2+b_2+2e_2=0.324 \text{ m. } [1.06 \text{ ft.}],$$

and hence the outer periphery of the pipe is

$$p_w=3.14\times 0.324\left(1+\frac{1}{4}\left(\frac{0.210}{0.324}\right)^2\right)=1.124 \text{ m. } [3.69 \text{ ft.}]$$

If the blast is to be heated from $t_1=15°$ C. [59° F.] to $t_2=300°$ C. [572° F.], the total length of the heating-surface must be

$$l_w=0.2(t_2-t_1)\frac{Q}{p_w}=0.2\times 285\frac{2}{1.124}=101.5 \text{ m. [333 ft.]},$$

so that each of the six coils of pipe exposes to the fire a length $\frac{101.5}{6}=16.9$ m. [55.5 ft.]. If each set of pipes is composed of six pieces placed one over another, each piece must have a length of $\frac{16.9}{6}=2.81$ m. [9.25 ft.]. As the pipes project through the masonry, they must be of a greater length, say 3.2 m. [10.5 ft.]; then the length of one series is 19.2 m. [63 ft.] and the rectified length of each of the five elbows is 1.6 m. [5.25 ft.]; the total length of each of the six series is therefore

$$l_2=19.2+5\times 1.6=27.2 \text{ m.}$$

Consequently the frictional resistance in the heating-pipes is

$$\zeta_2\frac{l_2}{d_2}=\zeta_2\frac{p_2l_2}{4F_2}=0.025\frac{31.23\times 27.2}{4}=5.31.$$

Moreover, if we assume the sum of the coefficients of resistance for the five elbows and two tees of each series of pipe to be

$$\Sigma(\zeta)=5\times 0.5+2\times 1.25=5,$$

we have

$$\Sigma(\zeta)+\zeta_2\frac{l_2}{d_2}=10.31.$$

Moreover,

$$x=\frac{1+0.00367\times 300}{1+0.00367\times 15}=1.99,$$

and

$$x^2-1=2.97,\quad 2x^2=7.94,\quad \frac{x^2+1}{2}=2.48.$$

The velocity of the air at the entrance to the pipes is, neglecting compression,

$$v_1 = 10 \text{ m. } [32.8 \text{ ft.}];$$

hence

$$\frac{v_1^2}{2g} = 5.1 \text{ m. } [16.7 \text{ ft.}]$$

and

$$\frac{v_1^2}{2g(b+h)} = \frac{5.1}{0.760+0.160} = 5.54 \text{ m. } [18.2 \text{ ft.}].$$

Furthermore, the density of mercury relatively to the outer air at 15° C. [59° F.] is

$$\varepsilon = \frac{13,600}{1.294}(1+0.00367\times 15) = 11,090;$$

then the lowering of the manometer for the passage through the heating-pipes is, according to (6),

$$y = \frac{x^2 - 1 + \left(\Sigma(\zeta) + \zeta_2 \frac{l_2}{d_2}\right)\frac{x^2+1}{2}}{\varepsilon - \left(2 + \Sigma(\zeta) + \zeta_2\frac{l_2}{d_2}\right)x^2 \frac{v_1^2}{2g(b+h)}} \frac{v_1^2}{2g}$$

$$= \frac{2.97 + 10.31\times 2.48}{11,090 - 12.31\times 3.97\times 5.54} 5.1 = 0.013 \text{ m. } [0.0426 \text{ ft.}].$$

Hence the height of the blast-manometer at the outlet of the heating apparatus is

$$z = h - y - \left(1 + \zeta_0 + \zeta_1\frac{l_1}{d_1}\right)\frac{v_1^2}{2g\varepsilon} = 0.160 - 0.013 - \left(1.5 + 0.025\frac{5}{0.5}\right)$$

$$\times \frac{10^2\times 0.051}{11,090} = 0.160 - 0.013 - 0.0008 = 0.146 \text{ m. } [0.48 \text{ ft.}],$$

and we have

$$1 + 0.028\frac{z}{b} = 1 + 0.028\frac{146}{760} = 1.0054.$$

Also, as in the example in the preceding article,

$$\left(\frac{F_u}{F_s}\right)^2=0.0034, \quad \zeta_u=0.1814,$$

and

$$\left(\Sigma(\zeta)+\zeta_s\frac{l_s}{d_s}\right)\left(\frac{F_u}{F_s}\right)^2=\left(2+0.025\frac{10}{0.292}\right)0.0034=0.010,$$

so that we have, according to (10),

$$w=1-0.0034+0.1814+0.01=1.188.$$

Finally, we find the total cross-section of the tuyeres, from (12), to be

$$\begin{aligned}F_u&=\frac{1+0.028\frac{z}{b}}{396(1+0.00367\times15)u}Q\sqrt{(1+0.00367\times300)w\frac{b}{z}}\\&=\frac{1.0054\times2}{396\times1.055\times0.92}\sqrt{2.10\times1.188\times\frac{760}{146}}\\&=0.0189 \text{ sq. m. } [0.203 \text{ sq. ft.}].\end{aligned}$$

Consequently each of the three tuyeres must have a cross-section of 0.0063 sq. m. [0.068 sq. ft.], which corresponds to a diameter of about 90 mm. [3.53 ins.]. In § 18 we found the diameter of a tuyere-orifice to be only 74 mm. [2.91 ins.] when a cold blast was employed.

§ 20. **Dimensions of Piston-blowers.**—To design a piston-blower we must know the quantity of blast Q to be delivered and the blast pressure or manometer height h; by blast pressure is ordinarily understood the excess of the pressure of the blast over that of the atmosphere. Let h represent the height of the manometer, and γ the density of the manometer fluid; then the blast pressure is expressed by

$$p_1-p=h\gamma.$$

With a mercury manometer and h expressed in meters [inches] we have for the blast pressure per square centimeter [per square inch]

$$p_1-p=1.36h \text{ kg. } [p_1-p=0.491h \text{ lbs.}],$$

and therefore

$$h=0.735(p_1-p) \text{ m. } [h=2.04(p_1-p) \text{ ins.}].$$

With a water-manometer we have

$p_1-p=0.1h$ kg. $[p_1-p=0.434h$ lbs. when h is expressed in ft.],

and inversely

$$h=10(p_1-p) \text{ m. } [h=2.31(p_1-p) \text{ ft.}].$$

In furnaces for producing copper we have

$$h=0.040 \text{ meter mercury } [1.57 \text{ ins.}],$$

or

$$p_1-p=0.054 \text{ kg. } [0.77 \text{ lb. per sq. in.}].$$

In furnaces for producing pig iron and burning charcoal we have

$$h=0.040 \text{ to } 0.065 \text{ m. } [1.57 \text{ to } 2.56 \text{ ins.}],$$

or

$$p_1-p=0.054 \text{ to } 0.088 \text{ kg. } [0.77 \text{ to } 1.25 \text{ lbs. per sq. in.}];$$

using light coke,

$$h=0.08 \text{ to } 0.13 \text{ m. } [3.15 \text{ to } 5.12 \text{ ins.}],$$

or

$$p_1-p=0.109 \text{ to } 0.18 \text{ kg. } [1.55 \text{ to } 2.56 \text{ lbs. per sq. in.}];$$

and using heavy coke or anthracite,

$$h=0.15 \text{ to } 0.18 \text{ m. } [5.91 \text{ to } 7.09 \text{ ins.}],$$

or

$$p_1-p=0.2 \text{ to } 0.25 \text{ kg. } [2.84 \text{ to } 3.56 \text{ lbs. per sq. in.}].$$

In blowers for *Bessemer plants* the excess of pressure is not more than one atmosphere [29.92 ins.], and the blowers which furnish air for pneumatic hoists require pressures of 4 to 6 atmospheres [60 to 90 lbs. per sq. in.].

The average volume of blast per minute for a copper-producing furnace is

$$60Q = 6 \text{ cu. m. } [212 \text{ cu. ft.}];$$

for an iron blast-furnace using charcoal it is

$$60Q = 15 \text{ to } 50 \text{ cu. m. } [530 \text{ to } 1770 \text{ cu. ft.}],$$

and when using coke

$$60Q = 60 \text{ to } 150 \text{ cu. m. } [2120 \text{ to } 5300 \text{ cu. ft.}].$$

If F is the piston area, s the piston stroke, and n the number of cylinder volumes of air delivered by the blower per minute, the theoretical volume of air is $Q_0 = \frac{nFs}{60}$; but as the actual volume of air is only 60 to 75 per cent. of the theoretical volume, we must place

$$Q = 0.60\frac{nFs}{60} \text{ to } 0.75\frac{nFs}{60};$$

or, if we represent the coefficient of discharge or the ratio (0.6 to 0.75) of the actual volume of the blast to its theoretical volume by ϕ, we have the general expression

$$Q = \phi\frac{nFs}{60},$$

and we also have

$$Q = \phi\frac{nzFs}{60}$$

when the blowing plant contains z *single-acting* pistons each making n double strokes per minute. On the other hand we have

$$Q = \phi\frac{nzFs}{30}$$

when the blowing plant contains z *double-acting* pistons each making n double strokes per minute.

Now $\frac{2ns}{60}=v$ is the average piston velocity; hence in the first case we must place

$$Q=\phi\frac{zFv}{2}$$

and in the second

$$Q=\phi zFv.$$

The average piston velocity is $v=0.6$ to 0.9 m. [2 to 3 ft.] in inefficient blowing apparatus, such as bellows and box-blowers, also exhausters and cylinder-blowers with narrow air-passages and small blast-pipes. With more perfect blowers and large blast-pipes v varies between 1.2 and 1.6 m. [4 to 5¼ ft.]

Finally, in slide-valve and clack-valve blowers, driven by high-pressure engines, the average piston speed varies from 2 to 3 m. [6 to 10 ft.].

From the assumed velocity v we obtain the sum of the piston areas

$$zF=\frac{2Q}{\phi v}$$

for *single*-acting, and

$$zF=\frac{Q}{\phi v}$$

for *double*-acting blowers.

The number z of blowing-cylinders which force air into a common reservoir varies from 2 to 4 when the pistons are single-acting, and from 1 to 2 when they are double-acting.

From z we can find the piston area F and the corresponding piston diameter

$$d=\sqrt{\frac{4F}{\pi}}=\sqrt{\frac{8Q}{\phi\pi zv}}=1.596\sqrt{\frac{Q}{\phi zv}}$$

for single-acting blowers and

$$d=\sqrt{\frac{4F}{\pi}}=\sqrt{\frac{4Q}{\phi\pi zv}}=1.128\sqrt{\frac{Q}{\phi zv}}$$

for double-acting blowers.

If we assume $\phi = 0.675$, we obtain for the first case

$$d = 1.942\sqrt{\frac{Q}{zv}},$$

and for the second case

$$d = 1.373\sqrt{\frac{Q}{zv}}.$$

In large cylinder-blowers which supply several coke-furnaces with air the piston diameter becomes 3 m. [10 ft.] and even more.

In piston-blowers with clack-valves the number of double strokes per minute is from 12 to 30, but in those with slide-valves the number is from 40 to 70. The latter are only suitable for low pressures because the loss of air increases with the pressure, as has been shown in § 13; therefore they are unsuitable for high furnaces, particularly those that produce pig iron by means of coke.

The cross-section of the suction and blast orifices should be proportional to the piston area and piston speed. In slowly running bellows and box blowers, where F is the piston area, the cross-section of the suction-orifices varies from $\frac{F}{15}$ to $\frac{F}{12}$; in cylinder-blowers of average speed this cross-section varies from $\frac{F}{10}$ to $\frac{F}{6}$; in rapidly running cylinder-blowers it varies from $\frac{F}{4}$ to $\frac{F}{2}$. The cross-sections of the blast-orifices can be considerably less; in slowly running blowers this cross-section $=\frac{F}{24}$ to $\frac{F}{18}$; in blowers of average speed it varies from $\frac{F}{16}$ to $\frac{F}{12}$. In very rapidly running blowers it may be $\frac{F}{8}$ to $\frac{F}{6}$. In order that the valves covering these orifices may open and close quickly they must be narrow and correspondingly numerous. They are provided with a ledge 10 to 25 mm. [0.4 to 1 in.] wide. In blowers with slide-valves the suction-orifices also act as blast-orifices;

their area is $\frac{F}{6}$ to $\frac{F}{10}$. The diameter of the blast-pipe is to be determined according to the rules given in § 18; in slowly running blowers the cross-section of the pipe is $\frac{F_v}{25}$ to $\frac{F_v}{20}$, and in those running with an average speed it is $\frac{F_v}{18}$ to $\frac{F_v}{12}$; in very rapidly running blowers it is $\frac{F_v}{10}$ to $\frac{F_v}{5}$, where F_v is the sum of the areas of the pistons that are blowing simultaneously.

If a blast-pipe conducts air of temperature t_2, its cross-section must be $\left(\frac{1+0.004t_2}{1+0.004t_1}\right)\left[\frac{1+0.002(t_2-32)}{1+0.002(t_1-32)}\right]$ times as large as the blast-pipe conducting the cold air of temperature t_1. This rule can also be employed to determine the sum of the cross-sections of the pipes in a hot-blast stove. The length of these pipes is to be determined according to § 19.

The *fuel* needed to generate the hot blast can be determined as follows: Each kilogram of coal burned gives out 6000 calories [1 lb. coal 10,800 B.T.U.], and this will heat, if the specific heat of air is taken equal to 0.25, $6000\times 4=24{,}000$ kg. of air about 1° C., or 80 kg$=\frac{80}{1.293}=62$ cu. m. of air about 300° C. [The combustion of 1 lb. of coal develops enough heat to raise 80 lbs. or $80\times 12.4=990$ cu. ft. of air about 540° F.] Consequently 1 cu. m. of air to be heated 300° C. requires $\frac{1}{62}=0.016$ kg. of coal [1 cu. ft. of air to be heated 540° F. requires about $\frac{1}{990}=0.001$ lb. of coal.] Experience shows that in hot-blast stoves only about 50 per cent. of the heat of combustion is utilized; therefore to heat 1 cu. m. of air there is needed about 30 g. of coal [to heat 1 cu. ft. of air about 0.002 lbs. of coal].

If we make the volume of air$=Q$ cu. m. and the fuel$=K$ kg., then for *lump* coal we have

$$K=0.03Q \quad [0.0019Q],$$

or the more general expression

$$K=0.0001(t_1-t)Q \text{ kgs.} \quad [K=0.0000035(t_1-t)Q \text{ lbs.}].$$

To determine the requisite cross-section of the tuyere orifices see § 18, etc.

The efficiency of a piston-blower, including the loss of blast due to the valves, etc., is

$$\eta = 0.40 \text{ to } 0.60,$$

which enables us to determine the work required per second of the engine, using for this purpose the formula (see § 10)

$$L = \left(1 - 0.3521\frac{h}{b} + 0.2000\left(\frac{h}{b}\right)^2\right)\frac{Qh\gamma}{\eta},$$

where b is the height of the barometer, h the height of the manometer on the reservoir, $Q = \frac{uFs}{60}$ the geometrically determined volume of the blast per second, and γ the density of the fluid used in the manometer.

Example.—In order to furnish a pair of blast-furnaces with $60Q = 120$ cu. m. [4238 cu. ft.] of air per minute with an excess of pressure of 0.2 kg. per sq. cm. [2.84 lbs. per sq. in.], a double-acting cylinder-blower is to be employed; required its dimensions and its other mechanical relations.

Let us assume the actual volume of the blast to be 67.5 per cent. of the geometrically determined volume, and the average piston speed $v = 1.5$ m. [4.92 ft.]; then the piston area must be

$$F = \frac{Q}{\phi v} = \frac{120}{0.675 \times 60 \times 1.5} = 1.975 \text{ sq. m. [21.26 sq. ft.]},$$

and the corresponding piston diameter is

$$d = \sqrt{\frac{4F}{\pi}} = 1.586 \text{ m. [62.44 ins.]},$$

and a stroke of 1.8 m. [5.9 ft.], the number of double strokes per minute is

$$n_2 = \frac{30v}{s} = \frac{30 \times 1.5}{1.8} = 25.$$

The cross-section of the suction-orifices can be assumed $\frac{F}{6} = \frac{1.975}{6} = 0.33$ sq. m. [3.55 sq. ft.].

With three square suction-orifices, like those in the blower represented in Fig. 14, the length of the side of one orifice becomes 0.35 m. [1.15 ft.], and in each orifice two clack-valves may be hung; but for the orifices of the delivery- or blast-valves the cross-section $\frac{F}{8}=\frac{1.975}{8}=0.24$ sq. m. [2.58 sq. ft.] will suffice, which can be obtained by two openings each 0.6 m. [1.97 ft.] long and 0.2 m. [0.66 ft.] wide.

The cross-section of a cold-blast pipe can be taken equal to $\frac{F_v}{10}=\frac{F}{10}=0.198$ sq. m. [2.13 ft.], and this gives a diameter of pipe of 0.5 m. [1.64 ft.] and a velocity of blast in it of 15 m. [49.2 ft.]. If the air is heated to 200° C. [400° F.] in the hot-blast stove, the heating-pipes and the hot-blast pipes must have a cross-section of $(1+0.004\times200)0.198=1.8\times0.198=$ 0.356 sq. m. [3.83 sq. ft.] and a diameter of 0.674 m. [2.21 ft.].

When the outer pressure of the atmosphere is 103 kg. per sq. cm. [14.65 lbs. per sq. in.] and therefore $\frac{h}{b}=\frac{0.2}{1.03}=0.1942$, so that $h\gamma=0.2$ kg. [2.84 lbs.] and the efficiency $\eta=0.5$, the power of the engine is

$$L=\left(1-0.3521\frac{h}{b}+0.2\left(\frac{h}{b}\right)^2\right)\frac{Fvh\gamma}{\eta}$$

$$=(1-0.3521\times0.1942+0.2\times0.0377)\frac{19,750\times1.5\times0.2}{0.5}$$

$$=(1-0.0683+0.0075)11,850=0.939\times11,850$$

$$=112 \text{ m.-kg.}=148.3 \text{ horse-power.}$$

§ 21. **Compressors.**—The extensive use of compressed air for running mining machinery and pneumatic railways, and in caissons and for marine purposes, has given rise to various machines for compressing air, and which are briefly termed *compressors*. In principle these machines are like cylinder-blowers, differing from them chiefly in imparting a higher pressure to the air. While blowers for blast-furnaces produce a blast having an excess of pressure of ¼ to ½ atmosphere, and those for Bessemer plants an excess of pressure of not more than 1½ to 2 atmospheres, the compressed air necessary to run

rock-drills has usually a pressure of 5 to 6 atmospheres; indeed there are exceptional cases in which the pressure of the air needs to be as high as 100 atmospheres, as, for instance, in the charging of torpedoes.

In consequence of the great compression, the air is considerably heated in all these machines, the heating increasing the farther the compression is carried. Since such heating of the compression cylinder not only injures the packing of the piston, stuffing-boxes, and valve, and renders lubrication difficult, but also considerably increases the resistance of the machines by increasing the pressure of the air, it is customary to cool the compression cylinders with water.

This cooling is effected either by putting on the cylinder a water-jacket, through which cold water is circulated, or by injecting into the cylinder water which is carried off with the air by the delivery-valves. This injected water has the advantage of filling up the hurtful space between the piston and valves, and thus diminishes the loss of volume due to this space.

It is clear that the disadvantage of the hurtful space increases with the compression of the air, for, during the return-stroke of the piston, the compressed air in the hurtful space must expand to the atmospheric pressure before new air can be sucked in. In *dry* compressors, i.e., such as work without injection, it is necessary to make the hurtful space as small as possible. In the older *wet* compressors water was not only injected during each stroke, but the space between the *piston* and *valves* was kept constantly filled with water. Such an arrangement was, however, accompanied by heavy shocks, particularly when there was a rapid change of stroke. Therefore in the later wet pumps it is preferred to inject only enough water to cool the cylinder and fill the *hurtful spaces*.

The pumps of compressors are either vertical or horizontal, single- or double-acting. The suction- and delivery-orifices in compressors are always closed by lift-valves, never by slide-valves, the reasons given in § 13 sufficiently indicating why high pressure of the air forbids the employment of slide-valves. It is only on a few recent and excellent high-speed compressors that arrangements have been made to give to these lift-valves a positive precise motion, so that their opening need not depend

on the existence of a sufficient vacuum in the cylinder. The diameter of the cylinder is seldom greater than 0.4 m. [16 ins.], the piston velocity is usually 1.2 to 1.5 m. [4 to 5 ft.], the number of double strokes is usually between 20 and 50, but there are constructions which permit more than 100 R.P.M. The lift of the valves must be as small as possible in all compressors, in order to obtain rapid action and consequently small losses of air. According to the velocity, the suction-valve openings are made from $\frac{1}{8}$ to $\frac{1}{10}$ of the cross-section of the cylinder, but the openings of the delivery-valve can be smaller.

Fig. 38 is a sketch of the compression-pumps employed in the construction of the Mont Cenis Tunnel * for the purpose of furnishing air of five atmospheres (gauge) pressure to the rock-drills. The two single-acting pumps are set horizontally and in line, each pump *A* being provided with a plunger-piston *B*, which is driven from the shaft *C* by means of the crank *CD* and a connecting-rod *DE*. The atmospheric air is drawn in

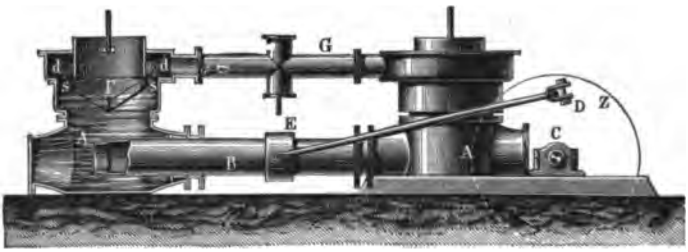

FIG. 38.

from *E*, passing through the suction-valve *s*, and is forced out of the delivery-valves *d* into the air-pipe *G*. The quantity of water in each cylinder is just sufficient to fill the space up to the delivery-valve *d* when the piston is at the end of its inward stroke, so that practically there is no hurtful space for the air and suction commences as soon as the piston begins its return-stroke. The crank-shaft *C* is driven by gearing set in motion by an overshot wheel. In the Gotthardt tunnel construction horizontal plunger-pumps were likewise employed, the plungers having 0.46 m. [18.11 ins.] diameter and 0.45 [17.71

* Zeitschr. f. Berg-, Hütten- und Salinenwesen, 1869; also *Rühlmann's* Allgemeine Maschinenlehre, Bd. IV.

ins.] stroke. Each group of three pumps was driven by a crank-shaft which received from a turbine 90 R.P.M., so that the piston velocity was 1.35 m. [4.43 ft.]. Three such pumps sucked in per minute 36.2 cu. m. [1278 cu. ft.] of air at atmospheric pressure, and this was compressed till it attained an excess of pressure of 7 atmospheres. The cooling in these pumps was effected by circulating water through the hollow pistons and through water-jackets completely surrounding each cylinder.

One of the best compressors with water injection is that of *Dubois* and *François* shown in Fig 39. Here the horizontal double-acting cylinder *A* has two inclined suction-valves *s* in its covers, and above these in each end of the cylinder is a disk-shaped delivery-valve *d*. Through the two jet-pipes *E*

FIG. 39.

flows a continuous stream of water, sufficient to cool the cylinder, the water in front of the piston at the end of the stroke filling the hurtful space. The excess of water is always pressed through the delivery-valve *d* into the air-conduit *G*, from which also starts the pipe which supplies the jet-cocks *E*, so that the water is forced into the cylinder by the pressure of the compressed air in the conduit. The figure shows how the water is used to make the valves *air-tight*. The designers give the following experimental data of the performance of this compressor having a piston diameter of 0.45 m. [17.71 ins.]. Here *V* represents the volume of air acted upon by the piston in

order to obtain 1 cu. m. of air of 5 atmospheres *absolute* pressure.

The tabular quantities opposite w are the coefficients of discharge, i.e., the ratios of the air actually drawn in, measured at atmospheric pressure, to the volume described in the same time by the piston.

We see from the table how greatly the losses of air increase with the number of revolutions.

RESULTS OF A DUBOIS & FRANÇOIS COMPRESSOR OF 0.45 M. [17.71 INS.] DIAMETER AND 1.2 M. [47.24 INS.] STROKE.

Revolutions per Minute.	15	20	25	30	35
Piston velocity........	0.60 m. [1.97 ft.]	0.8 m. [2.62 ft.]	1.0 m. [3.28 ft.]	1.20 m. [3.94 ft.]	1.40 m. [4.59 ft.]
Coefficient of discharge, w	0.94	0.92	0.90	0.86	0.78
Theoretical volume, V..	5.32 cu. m. [187.9 cu. ft.]	5.43 cu. m. [191.7 cu. ft.]	5.55 cu. m. [196.2 cu. ft.]	5.814 cu. m. [205.3 cu. ft.]	6.41 cu. m. [226.4 cu. ft.]
Actual discharge.....	5 cu. m.	5 cu. m.	5 cu. m.	5 cu. m.	5 cu. m.

In the light of these results the designers recommend a velocity of 30 R.P.M. or a piston velocity of 1.2 m. [4 ft.]. For greater velocities they employ a modified design whose essential features are shown in Fig. 40. As regards the delivery-valves, cylinder, piston, and injector, this arrangement agrees perfectly with that just described, but differs in the arrangement of the suction-valves s_1 and s_2. These are disk-shaped and are so connected by a horizontal rod s_0 that the opening of one causes the other to close. For example, if we suppose the piston B to be in the extreme right-hand position shown and on the point of moving to the left, then, as soon as motion begins, there will be a vacuum in the space D now filled with water, in consequence of which the valve s_1 will open and, by means of the connecting-rod, will cause the other valve, s_2, to close. The knife-edges p_1 and p_2 allow the valve-rod s_0 to move easily. By this arrangement the suction-valve is fully opened at the beginning of the piston stroke and needs no vacuum in the cylinder to enable the atmospheric pressure to open the valve. We see from this that these compressors, like blowers

with slide-valves, can be run faster than the ordinary blowers with clack-valves. The same problem of attaining a greater precision of opening the suction-valve has been solved in a

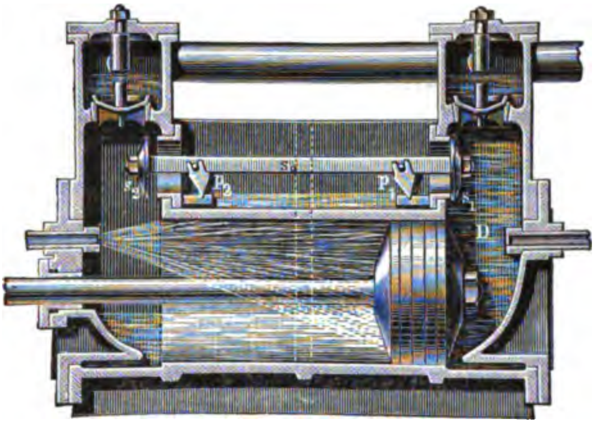

FIG. 40.

simple manner by *Sturgeon* of Leeds. Here the stuffing-boxes for the piston-rod which passes through both ends of the cylinder are modified so as to also act as suction-valves. The manner in which this is done is shown in Fig. 41,

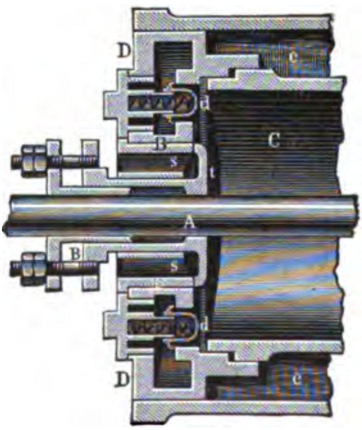

FIG. 41.

which represents a section of one end of the compression cylinder. The piston-rod A passes through a stuffing-box B which has a slight adjustment in the cover D of the cylinder. Here

the disk-shaped piece t serves to close the ring-shaped suction-opening s in the cylinder-cover. It is easy to see that at each reversal of the stroke the friction between the piston-rod A and the stuffing-box B causes the disk t to move with the piston-rod, thus closing the suction-valve at one end of the cylinder and opening it at the other end. No shocks of any consequence have been noticed in this machine, for the velocity of the piston-rod near the dead-centers is small even with a great number of revolutions.

The delivery-valves d in this design are likewise placed in the cylinder-covers D and arranged concentrically about the piston-rod, the air-conduit being directly attached to the hollow cover. The compressor is a dry one, the cooling being effected by water circulation in the jacket C. These machines can make from 150 to 200 R.P.M., and have recently been very extensively used.

Of the many other compressors we will only mention that of *Burleigh;* it is a representative of the vertical arrangement which has come into extensive use, particularly in the United States. The air-pumps are here single-acting, two being always employed, and driven by cranks diametrically opposite. A section of such a pump is shown in Fig. 42. The cylinder A is open below, the hollow plunger B being driven by the crank-shaft C and connecting-rod D. The piston-head carries the suction-valve s, while the delivery-valve d shuts off the air-chamber W which is connected with the air-pipe L. During the down-stroke of the piston, water is injected through the pipe s which, toward the end of the forcing action, fills the hurtful space between the piston and delivery-valve. The valves have a larger area, consequently a small lift is sufficient, and in the arrangement given the valves are rendered easily accessible by the removal of the cover E. These machines are usually driven by a vertical steam-engine, a third crank being placed for this purpose on the driving-shaft C. In order to equalize the motion as much as possible, the crank of the engine is usually so set relatively to the cranks of the compressor that when the steam-piston has traveled about one-eighth of its stroke the compressor-pistons will be at the dead-points, the full pressure of the steam acting when there is the greatest resistance in the compressors. In the experiments

made, the machine after running awhile at 60 revolutions per minute developed a temperature of 35° to 40° C. [95° to 104° F.].

To judge of the hurtful influence of the heating of the air

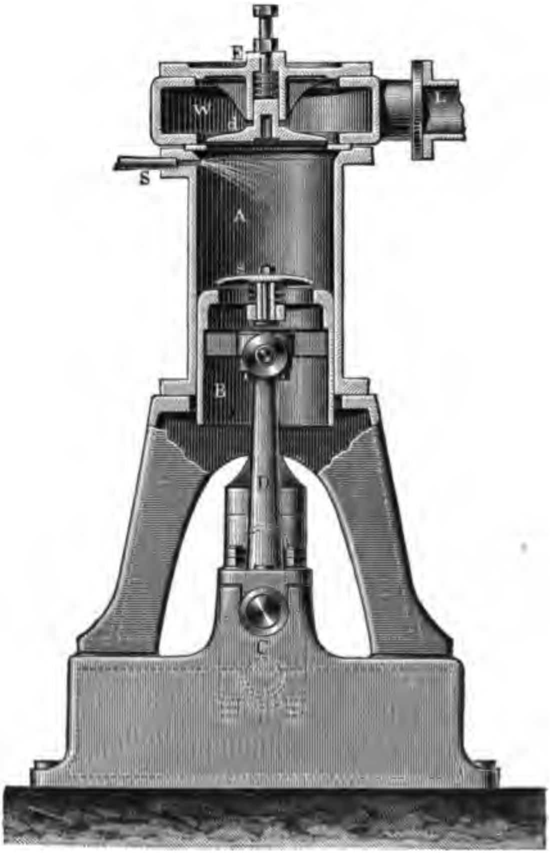

FIG. 42.

caused by compression, we may use the diagram Fig. 43. In it AB represents the length $l = 1.2$ m. [3.94 ft.] of the stroke of a *Dubois* and *François* compressor of 0.45 m. [17.71 ins.] diameter; the indicator-curve AE shows the increase of pressure of the air while it is being compressed from the atmospheric pressure AC to the pressure $DF = 5$ atmospheres.

Evidently the area $AEFBA$ is a measure of the work expended in this compression and in forcing the air out of the

cylinder. There are also in the figure two other curves AE_1 and AE_2 which were found by calculation, AE_1 corresponding to *Mariotte's law*, i.e., to the assumption that the temperature of the air remains constant (isothermal curve), while AE_2 was constructed on the supposition that no heat is lost by cooling (adiabatic curve). From this we see that in a machine which is not cooled at all the necessary work is greater by the area AE_2E_1 than in a machine in which the cooling off is perfect, i.e., in which the temperature remains constant, the work in the latter case being represented by the area AE_1FBA. Meas-

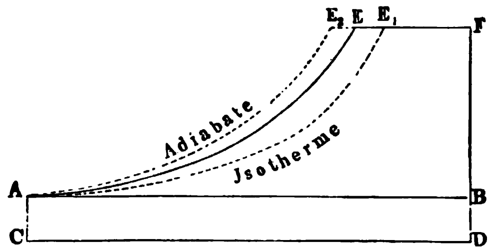

FIG. 43.

urement of these areas shows that for a compression from 1 to 5 atmospheres there is a loss of work of about 23 per cent. caused by the heating. It is now clear that the loss of work is smaller the more perfect the cooling, i.e., the more the indicator-curve AE approaches the isothermal line AE_1. This cannot of course be completely attained; but the economical value of the cooling in any existing machine can be estimated from the relative size of the area AE_2E included between the indicator and adiabatic curves.

§ 22. **The Driving Force of Blowers.**—The losses of power occasioned by the friction at the crank, at the guides, by the toothed wheels, etc., are to be determined by the formulas given in the Mechanics of Machinery, Part I, section 1. Let η_1 be the efficiency of the blower itself, η_2 that of the prime mover which is to be determined according to the formulas given in Vol. II, and η_3 the efficiency of the intermediate machinery. Then the efficiency of the whole blowing plant is $\eta = \eta_1\eta_2\eta_3$.

Let the prime mover be a *water-wheel*, Q_1 the quantity of water which enters it per second, h_1 the head and γ_1 the density of the water; also let Q be the quantity of blast required, meas-

ured at the outer pressure and temperature, h the height of the manometer, and γ the density of the manometer fluid; then, according to a preceding article, we have

$$\eta Q_1 h_1 \gamma_1 = \left(1 - 0.3521\frac{h}{b} + 0.2\left(\frac{h}{b}\right)^2\right) Q h \gamma,$$

and from this we can determine for a given volume of blast of given pressure the quantity of water needed by the wheel, namely,

$$Q_1 = \left(1 - 0.3521\frac{h}{b} + 0.2\left(\frac{h}{b}\right)^2\right) \frac{Q h \gamma}{\eta h_1 \gamma_1};$$

moreover,

$$Q = x\frac{zn}{60}Fs = xz\frac{Fv}{2}$$

or

$$= x\frac{zn}{30}Fs = xzFv$$

according as the blower is composed of z single-acting or z double-acting pistons, x representing the coefficient of discharge (see § 18), F the piston area, s the piston stroke, v the piston velocity, and n the number of double strokes of a blower per minute. From the given number of revolutions u of the water-wheel we can now find the required velocity ratio of a train of gearing, according to

$$\phi = \frac{n}{u} = \frac{r_1}{r},$$

and the necessary determinations, with respect to the radii r and r_1 and the number of teeth of the gears, can now be made. The necessary dimensions and relations of the water-wheel can of course be determined from Q_1 and u (see Vol. II).

If a steam-engine drives the piston-blower, the arrangement and calculation of the whole blowing plant is simpler, because generally there is a direct transmission of the steam force to the blower. Here the efficiency of the blowing-engine is $\eta = \eta_1 \eta_2$, i.e., a product of the efficiency of the steam-engine η_2 and the

efficiency η_1 of the blower. We may therefore use the simpler formula

$$\eta Q_1 p\left(1+\log_e \varepsilon-\frac{q}{p_1}\right)=\left(1-0.3521\frac{h}{b}+0.2\left(\frac{h}{b}\right)^2\right)Qh\gamma,$$

where Q_1 is the quantity of steam furnished to the engine, p the boiler pressure, q the pressure in the condenser or of the atmosphere as the case may be, ε the ratio of expansion, and $p_1=\frac{p}{\varepsilon}$ the calculated steam pressure at the end of the stroke. Accordingly the quantity of steam needed to produce a given volume of blast Q and blast pressure h is

$$Q_1=\frac{1-0.3521\frac{h}{b}+0.2\left(\frac{h}{b}\right)^2}{1-\log_e \varepsilon-\frac{\varepsilon q}{p}}\frac{Qh\gamma}{\eta p},$$

and from this we can determine, with the help of the directions given in Vol. II, the necessary relations, dimensions, etc., of the steam-engine. Of course in direct-acting steam-engine blowers the number of double strokes, velocity, etc., of the engine is the same as that of the blowers.

Example.—A blower is to deliver to a blast-furnace 60 cu. m. [2019 cu. ft.] of air per minute with an excess of pressure of 100 mm. [3.94 ins.], and is to be driven by a water-power having a head of 8 m. [26.24 ft.]; how much water is needed? If we assume the efficiency of the blower to be $\eta_1=0.60$, that of the intermediate machinery to be $\eta_2=0.90$, and that of the water-wheel to be $\eta_3=0.75$, the efficiency of the whole plant is

$$\eta=\eta_1\eta_2\eta_3=0.60\times0.90\times0.75=0.405;$$

and if we assume the smallest height of the barometer for the locality to be $h=740$ mm. [29.13 ins.], then the required quantity of water per second is

$$Q_1=\left(1-0.3521\frac{h}{b}+0.2\left(\frac{h}{b}\right)^2\right)\frac{Qh\gamma}{\eta h_1\gamma_1}$$

$$=\left(1-0.3521\frac{10}{74}+0.2\left(\frac{10}{74}\right)^2\right)\frac{60\times0.1\times13.6}{60\times0.405\times8}=0.401 \text{ cu. m.}$$

[14.16 cu. ft.]

If the blower consists of two double-acting pistons working with the average velocity $v=1$ m. [3.28 ft.], then, with a coefficient of discharge $x=0.65$, the required piston area is

$$F=\frac{Q}{xzv}=\frac{60}{0.65\times2\times1\times60}=0.769 \text{ sq. m. [8.3 sq. ft.]},$$

and accordingly the piston diameter is

$$d=0.989=\text{(nearly) } 1 \text{ m. [3.28 ft.]}.$$

If we give to each piston a stroke $s=1.2$ m. [3.94 ft.], the number of double strokes or revolutions per minute of the crank-shaft is

$$n=\frac{30v}{s}=\frac{30}{1.2}=25.$$

On the other hand, if the water-wheel makes 6 revolutions per minute, we must employ a pair of gears having the velocity ratio of $\frac{25}{6}$, i.e., we must place on the wheel-shaft a toothed gear with $\frac{25}{6}$ as many teeth as the gear on the crank-shaft. For example, if the number of teeth of the latter equals 24, the number of teeth of the former must be 100.

§ 23. **The Fly-wheels of Cylinder-blowers.**—An important matter in designing a cylinder-blower is the determination of the size of the fly-wheel which is to impart the desired degree of uniformity to the motion. In blowers, particularly vertical ones, the weights of the pistons, cranks, etc., must of course be balanced by counterweights, and in this connection reference may be made to Vol. III, 1. The variation in the velocity of blowers is due not only to the variation of the steam pressure resulting from the expansion, but also to the variable resistance offered by the air to the blower-piston. This last variation increases with the blast pressure, and is therefore greater in Bessemer blowers than in ordinary blowers for blast-furnaces,

and is greatest in compressors. Moreover, in determining the size of the fly-wheel, it makes considerable difference whether the blower is a direct-acting one, in which steam- and blower-pistons are fastened to the same piston-rod, or whether the blower-piston is driven by a crank-shaft which is turned by the engine by means of a special crank-train.

Let us first assume the simpler case of a *direct-acting* blowing-engine, then we easily see that with a small blast pressure, such as is usual in blowers for blast-furnaces, the uniformity of motion is greater than in an ordinary steam-engine, other things being equal, i.e., with equal cylinder and fly-wheel dimensions and equal ratios of expansion. The correctness of this remark follows from the supposition of a small blast pressure, the *resistance of the blower-piston* being then nearly constant during the whole stroke, and consuming at every instant a nearly constant portion of the driving force of the steam-piston. But this is not the case in an ordinary steam-engine with a constant *circumferential* pressure at the crank, for we know from the theory of the crank-train (see Vol. III, I) that a constant circumferential resistance of the crank calls forth very variable resistances against the piston.

For example, if the steam-engine worked without expansion and the piston resistance were constant, the fly-wheel would only be needed on account of the inertia of the reciprocating masses and the variable frictional resistances of the crank-train. But the matter is very different with greater blast pressure, and it is easy to see that, as the period of *greatest resistance* of the blower-piston is the same as that of *smallest steam pressure,* the causes of variable motion exist, generally speaking, in a higher degree than in an ordinary steam-engine. The investigation must therefore be made separately for each case, and for this purpose it is best to employ the graphical method on account of its clearness and simplicity, the analytical calculations being quite complex. This last is particularly true of indirect-acting engines, and we will therefore undertake the calculation only for the simplest case of a direct-acting engine.

The required rotating mass may now be determined in the manner indicated in Vol. III, I. While the piston has traveled from the end of its stroke any distance x_1. the crank has turned from the dead-point through the angle α, the velocity v_1 of the

crank-pin at the dead-point being changed to v. If ε represents the ratio of expansion of the steam-engine, i.e., if

$$s'=\frac{s}{\varepsilon}=\frac{2r}{\varepsilon}$$

is the travel of the piston before the steam is cut off, and if $P=Fp$ is the steam pressure on the piston during the admission period, and R the back pressure, the work performed by the piston while traversing the distance x is given by

$$A_1=Ps'\left(1+\log_e\frac{x_1}{s'}\right)-Rx_1$$

$$=Ps'\left(1+\log_e\frac{(1-\cos\alpha)\varepsilon}{2}\right)-Rr(1-\cos\alpha), \quad . \quad (1)$$

where $x=r(1-\cos\alpha)$ is introduced on the supposition that the connecting-rod is long.

Moreover, if we assume the excess of pressure on the piston, after compression is completed, to be $F_1h\gamma=Q$, and let s_0 represent the distance which the piston must travel before the air can be brought to the excess of pressure $h\gamma$, then the work performed by the piston is given by

$$A_2=Q\left(\frac{s_0}{2}+x_1-s_0\right)=Q\left(r(1-\cos\alpha)-\frac{s_0}{2}\right). \quad . \quad . \quad (2)$$

It will be sufficiently accurate to determine s_0 from

$$\frac{b+h}{b}=1+\frac{h}{b}=\left(\frac{s}{s-s_0}\right)^x=1+x\frac{s_0}{s},$$

or

$$s_0=\frac{h}{xb}s.$$

The excess A_1-A_2 of the work done by the steam over the useful work has been expended in accelerating the masses; and therefore if m_1 represents the rotating mass reduced to the

crank-pin and m_2 the reciprocating mass, we have, as before (see Vol. III, I),

$$A_1-A_2=Ps'\left(1+\log_e\frac{(1-\cos\alpha)\varepsilon}{2}\right)$$

$$-Q\left(r(1-\cos\alpha)-\frac{s_0}{2}\right)-Rr(1-\cos\alpha)$$

$$=m_1\frac{v^2-v_1^2}{2}+m_2\frac{v^2}{2}\sin^2\alpha. \qquad (3)$$

From this follows

$$v^2=\frac{m_1v_1^2+2(A-A_2)}{m_1+m_2\sin^2\alpha}; \qquad (4)$$

at all events m_1 will be so large that v will differ but slightly from v_1, and we can therefore write approximately

$$v=v_1\left(1+\frac{A_1-A_2}{m_1v_1^2}-\frac{m_2}{2m_1}\sin^2\alpha\right),$$

or, after substituting the values of A_1 and A_2,

$$v=v_1+\frac{Ps'\left(1+\log_e\frac{(1-\cos\alpha)\varepsilon}{2}\right)-(Q+R)r(1-\cos\alpha)+Q\frac{s_0}{2}}{m_1v_1}$$

$$-\frac{m_2v_1}{2m_1}\sin^2\alpha.. \qquad (5)$$

The greatest and least velocity can be obtained by placing $\frac{\delta v}{\delta\alpha}=0$, thus obtaining

$$Ps'\frac{\sin\alpha}{1-\cos\alpha}-(Q+R)r\sin\alpha-m_2v_1^2\sin\alpha\cos\alpha=0. \qquad (6)$$

This equation is satisfied by $\sin\alpha=0$, i.e., for the dead-point, at which the velocity is a minimum.

To determine the crank-angle for the maximum velocity, transform equation (6) into

$$m_2v_1^2\cos^2\alpha+[(Q+R)r-m_2v_1^2]\cos\alpha=(Q+R)r-Ps'. \qquad (7)$$

From this we find

$$\cos\alpha = -\frac{(Q+R)r - m_2 v_1^2}{2m_2 v_1^2}$$
$$+\sqrt{\frac{(Q+R)r-Ps'}{m_2 v_1^2}+\left(\frac{(Q+R)r-m_2 v_1^2)}{2m_2 v_1^2}\right)^2}. \quad . \quad (8)$$

If we substitute these values $v_{max.}$ and $v_{min.}$ in the known formula $v_{max.} - v_{min.} = \delta v$, where δ is the permissible coefficient of fluctuation, we obtain an equation for determining the mass m_1 of the fly-wheel.

For $\alpha = 180°$ we have of course the velocity $v = v_1$, so that for this case $A_1 = A_2$, that is,

$$Ps'(1+\log_e \varepsilon) - R2r = Q\left(2r - \frac{s_0}{2}\right).$$

By introducing the piston areas F and F_1 and placing $R = Fp_0$, where p_0 represents the atmospheric pressure, we get

$$Fp \times 2r\left(\frac{1+\log_e \varepsilon}{\varepsilon} - \frac{p_0}{p}\right) = F_1 h\gamma\left(2r - \frac{s_0}{2}\right),$$

or

$$P\left(\frac{1+\log_e \varepsilon}{\varepsilon} - \frac{p_0}{p}\right) = Q\left(1 - \frac{s_0}{4r}\right). \quad . \quad . \quad . \quad . \quad (9)$$

Example.—Required the weight of a fly-wheel for a direct-acting blowing-engine having a stroke of 1 m. [3.28 ft.] and a blowing-cylinder 1.2 m. [3.94 ft.] in diameter, the steam cutting off at $\frac{1}{3}$ stroke and the blower generating a blast having an excess of pressure of $\frac{1}{5}$ atmosphere.

Here $\dfrac{h}{b} = 0.2$, and therefore the travel of the piston when the blowing begins (i.e., at the end of compression) is

$$s_0 = \frac{h}{1.42b}s = 0.1408s = 0.1408 \text{ m. } [0.46 \text{ ft.}].$$

The stroke of piston during admission of the steam is

$$s' = \frac{s}{\varepsilon} = \frac{1}{3}s = \frac{1}{3} \text{ m. [1.093 ft.]}.$$

If we assume the absolute steam pressure in the cylinder to be $p=5$ atmospheres, and take the frictional resistances of the two pistons into account by taking the back pressure on the steam-piston to be $p_0 = 1.5$ atmospheres, we get, from equation (9),

$$\frac{P}{Q} = \frac{1-\frac{s_0}{4r}}{\frac{1+\log_e \varepsilon}{\varepsilon} - \frac{p_0}{p}} = \frac{1-0.0704}{\frac{1}{3}(1+1.0986)-0.3} = \frac{0.9296}{0.399} = 2.33.$$

Hence $P = 2.33Q$ and the back pressure

$$R = \frac{1.5}{5}P = 0.3P = 0.7Q.$$

The resistance Q becomes

$$Q = F_1 h\gamma = \frac{3.14 \times 1.2^2}{4} \times \frac{1}{5} 0.75 \times 13{,}600 = 1.131 \times 2040 =$$
$$= \text{(nearly) } 2300 \text{ m.-kg. [16,690 ft.-lbs.]}.$$

Now if we suppose the weight of the two pistons, the piston-rod, the cross-head, and the part of the connecting-rod that is reduced to the cross-head to be together equal to 1200 kg. [2646 lbs.], we have

$$m_2 = \frac{1200}{9.81} = 122.3 \text{ [82.17]}.$$

If the machine makes 36 R.P.M., the average velocity of the crank is

$$v_1 = \frac{36 \times \pi \times 1}{60} = 1.885 \text{ m. [6.18 ft.] per sec.},$$

and therefore

$$m_2 v_1^2 = 122.3 \times 1.885^2 = 435 \text{ m. [3145 ft.-lbs.]}.$$

With these values of P, Q, R, and $m_2v_1^2$, we obtain from equation (8) the angle α corresponding to the greatest velocity $v_{\text{max.}}$, namely,

$$\cos\alpha = -\frac{1.7\times2300\times0.5-435}{2\times435}$$

$$+\sqrt{\frac{1.7\times2300\times0.5-2.33\times2300\times\frac{1}{3}}{435}+\left(\frac{1.7\times2300\times0.5-435}{2\times435}\right)^2}$$

$$= -1.747+\sqrt{0.3885+3.0520}=0.108,$$

which gives $\alpha = 83° 48'$.

With this value of α we can now determine from (1) the work of the steam-piston:

$$A_1 = 2.33\times Q\times\tfrac{1}{3}\left(1+\log_e\frac{0.892\times3}{2}\right)-0.7\times Q\times0.5\times0.892$$

$$=(0.777\times1.2902-0.3122)Q=0.690Q,$$

and in like manner from (2) the useful work,

$$A_2 = Q(0.5\times0.892-0.0704)=0.376Q.$$

Accordingly the work expended in acceleration is

$$A_1-A_2=(0.69-0.376)Q=0.314Q=722 \text{ m.-kg. } [5222 \text{ ft.-lbs.}].$$

Moreover, since $\sin^2\alpha=\sin^2 83° 48'=0.984$, we have from equation (5)

$$v_{\text{max.}}=v_1\left(1+\frac{A_1-A_2}{m_1v_1^2}-\frac{m_2}{2m_1}\sin^2\alpha\right)=v_1\left(1+\frac{722}{m_1v_1^2}-\frac{122.3}{2m_1}0.984\right),$$

from which we get

$$\frac{v_{\text{max.}}}{v_1}-1=\frac{722}{m_1v_1^2}-\frac{60.2}{m_1}.$$

Now if the machine is to run with a degree of fluctuation $\delta=\frac{1}{30}$, we have

$$\delta=\frac{v_{\text{max.}}-v_1}{v_1}=\frac{v_{\text{max.}}}{v}-1=\frac{722}{m_1v_1}-\frac{60.2}{m_1},$$

and from this

$$m_1 = \frac{1}{\delta}\left(\frac{722}{1.885^2} - 60.2\right) = 30 \times 143.2 = 4296. \quad [2883.]$$

This is the mass reduced to the crank-pin circle. If we assume the radius of the fly-wheel to be four times as great as that of the crank, we obtain for the weight of the fly-wheel rim

$$G = \frac{m_1}{16} g = \frac{4296}{16} 9.81 = 2633 \text{ kg. } [5805 \text{ lbs.}].$$

In order to obtain an idea of the forces which prevent a direct-acting blowing-machine from running with a uniform motion, we shall make use of a diagram.

Let the base AB, Fig. 44, be equal to the stroke $2r$, and at each piston position erect two ordinates, one representing the pressure of the air on the blower-piston. This construction gives us the line $GHMJ$, which shows the variations of steam pressure when the steam is cut off at piston position H_0. The straight line CD, drawn parallel to the base at the distance AC, represents the constant back pressure on the other side of the piston, i.e., it represents the pressure of the atmosphere or of the condenser, as the case may be. Therefore if F represents the area of the steam-piston, p the driving steam pressure, and p_0 the back pressure of the atmosphere (or of the condenser), we have

$$AG = Fp \quad \text{and} \quad AC = Fp_0;$$

and assuming *Mariotte's law*, we shall find

$$BJ = \frac{AH_0}{AB} AG.$$

In like manner for a blower-piston which has the cross-section F_1 and compresses the air from the atmospheric pressure p_0 up to p_1, we find

$$AC_1 = F_1 p_0 \quad \text{and} \quad BK_1 = F_1 p_1,$$

and thus get the curve C_1E_1, representing the increase of pressure of the air. This curve can be constructed as an isothermal

curve when the heating of the air due to compression is diminished by the employment of the cooling methods; in case no cooling is attempted we must employ the formula

$$\frac{p_1}{p_0} = \left(\frac{V_0}{V_1}\right)^x$$

and draw the adiabatic curve C_1E_1 accordingly.

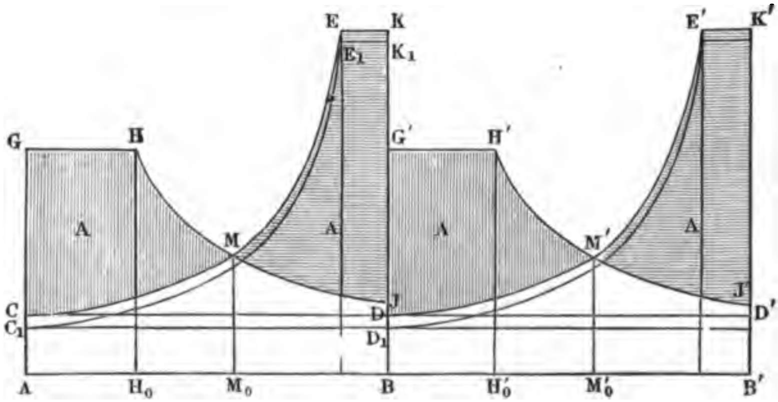

FIG. 44.

It follows that the work of the steam for each stroke is represented by the area $C_1E_1K_1D_1C_1$. Now if the latter area be lifted by the amount C_1C, so that it stands on the same base as the area of effective steam pressures, it will take the position $CEKDC$, and the intersection M of the two curves will have a projection M_0, which is evidently the piston position corresponding to the greatest velocity v_1 of the engine. For in this position the effective steam pressure is just equal to the resistance of the blower-piston; while the piston is traversing the distance AM_0 there is an excess of driving force, consequently acceleration takes place, and during the rest of the piston-stroke M_0B the excess of resistance retards the crank. During the return-stroke the procedure is repeated in exactly the same manner, and we obtain for this return-stroke a diagram like that for the forward stroke.

In the figure the diagram for the return-stroke is drawn beside that for the forward stroke, being placed on the base $BB' = 2r$. We see from this that on passing the dead-point B

an excess of driving force $DG' = CG$ suddenly occurs, and conclude that the smallest crank velocity $v_{\text{min.}}$ must take place at the dead-points. From what was said in Vol. III, I, concerning fly-wheels, it is plain that the variation of velocity experienced by the crank-pin may be found from the equation

$$m\frac{v_1^2 - v_2^2}{2} = A,$$

$$v_1 - v_2 = qv$$

and

$$v = \frac{v_1 + v_2}{2} \text{ approximately,}$$

where q is the permissible coefficient of fluctuation;

$$A = qv^2$$

where m represents the moving mass reduced to the crank-pin, and A the work measured by either of the equal shaded areas $CGHMC$ or $JKEMJ$, which do *not* lap in the two diagrams. How the dimensions of the fly-wheel can be obtained from the given degree of fluctuation was explained in Vol. III, I, in connection with fly-wheels.

The diagram here drawn for a direct-acting blowing-engine would also answer for an indirect-acting engine in which the steam-piston acted on a special crank on the driving-shaft, provided this crank was placed in the same direction as that of the double-acting blowing-piston, so that the two pistons would reach their dead-points at the same time. The diagram would also hold if the two cranks of the steam- and blowing-cylinders had unequal lengths r and r_1, for it would only be necessary to reduce the ordinates representing the pressures in the ratio $r : r_1$ in order to lay off both diagrams on the same line.

But if the crank of the steam-engine makes an angle with that of the blower, as is always the case in practice, the figure changes, the steam diagram $AGHJB$ shifting, relatively to the air-pressure diagram $AC_1E_1K_1B$, an amount depending on the angle formed by the two cranks. For instance, if the cranks are at right angles, the blower-piston will be nearly at

the middle of its stroke when the steam-piston reaches its dead-point. In this case, therefore, the initial ordinate AG of the steam-pressure diagram must be laid off at this middle position of the blower-piston. This shows that the graphical method will enable us to determine that angle included by the cranks for which the motion of the machine is most uniform.

For this purpose it is only necessary to bring the steam diagram into a position which will enable it to cover as much as possible of the air diagram, for then the work expended in acceleration or retardation is relatively smallest, for we know from what has preceded that these quantities of work are proportional to the areas that do not lap in the two diagrams. Such a diagram has been drawn in Fig. 45, and we at once see from it that, with an arrangement of cranks corresponding to this figure, the motion of the machine will be comparatively uniform. The two diagrams here cut each other in eight points

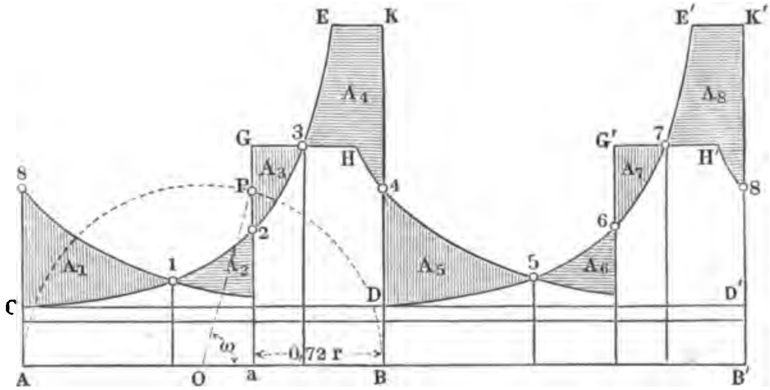

FIG. 45.

1, 2, 3 . . . 8, so that they project beyond each other in eight small areas A_1, $A_2 \ldots A_8$. These areas alternately represent excessive driving forces or accelerations and excessive resistances or retardations, the accelerations A_1, A_3, A_5 and A_7 having vertical hatchings in the figure, while A_2, A_4, A_6, and A_8, which indicate retardation, have horizontal hatchings. Of course the intersections 1, 3, 5, and 7 correspond to maximum velocities of the crank, while 2. 4, 6, and 8 correspond to the minimum velocities. The relative position of the cranks can be readily found, for the blower-piston must have traveled a distance Aa when the steam-piston

reaches a dead-point; in the figure $aB=0.36$, $AB=0.72r$; this distance therefore corresponds to an angle of advance of the blower-crank relatively to the steam-crank which is represented by w and found from $0.72r=r(1-\cos w)$ to be $w=79° 30'$. This diagram also corresponds to *Burleigh's* air-compressor shown in Fig. 42, for it is of no consequence here whether the main shaft drives a double-acting blower with one crank, or two equal single-acting pumps with two cranks which are diametrically opposite.

In the foregoing discussion, in order that the figure might be as simple as possible, no attention was paid to the inertia of the reciprocating masses; but when the machine runs with great velocity the influence of these masses cannot be neglected, and then the diagram may be drawn without difficulty, as indicated in Vol. III, I.

§ 24. **Rotary Blowers.**—Recently blowers have been constructed with *oscillating cylinders* resembling oscillating steam-engines (see Vol. II), by *Jolly* and by *Robert* (see Armengaud's Publication industrielle, Vol. XII).

But *rotary-piston blowers* are more extensively used; they are provided, *like rotary pumps* (see Mechanics of Pumping Machinery), with rotating pistons; such machines are principally used as exhausters, but are only used with advantage where small pressures are to be generated.

Under this head comes first the ventilator or exhauster of *M. Fabry*. This consists of two shafts C and D, Fig. 46, which have three pairs of main arms CA_1, CA_2, CA_3, and DB_1, DB_2, and DB_3, which carry large floats and are provided with cross-arms $E_1F_1, G_1H_1, \ldots$ whose epicycloidal ends, for example E_1 and H_1 as well as F_1 and G_2, act on each other like the teeth of two toothed wheels. These wheels hang in two troughs K and L to which the air is admitted by the passage M, and outside of the troughs the wheels are driven by two equal-toothed gears R and S, so that they will run with equal speed in opposite directions when one of the gears is driven by an outside force, say a steam-engine.

The troughs K and L are of masonry and lined with cement, which is put on after the wheels are hung in order that they may be as air-tight as possible. The floats fastened to each pair of main arms are made of boards or of sheet iron, and of course

occupy almost the whole length, 2 m. [6.5 ft.], of the trough. In order that the space DE_1F_1 between the wheels may be as air-tight as possible, it is necessary to connect each pair of cross-arms with sheet-iron or wooden floats, and these, as well as the epicycloidal floats at the ends of the arms, must extend from one end of the trough to the other. When the machine turns,

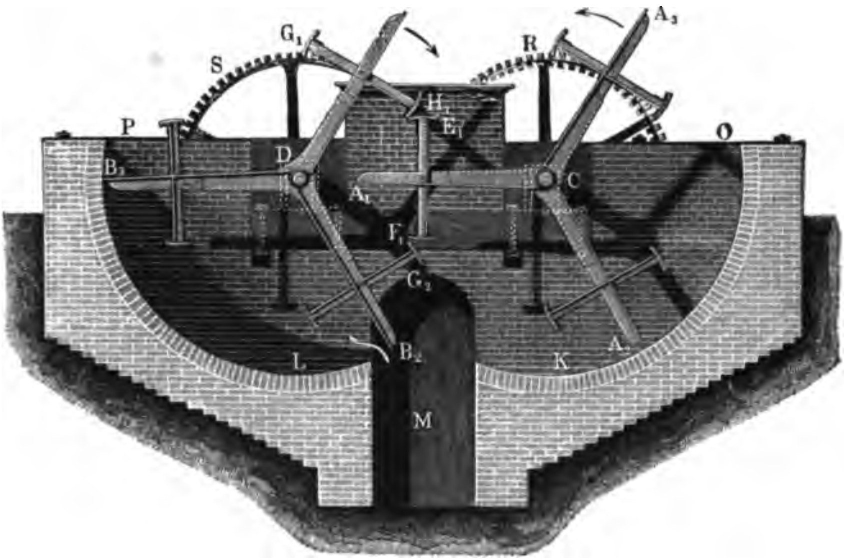

FIG. 46.

the floats on the main arms draw in air from M and discharge it at O and P near the outer circumferences. On the other hand the space DE_1G_2, between the cross-arms in contact, carries back a certain volume of air from the atmosphere to the space M; the actual discharge is therefore the difference between these two quantities of air, and will be determined hereafter. In order to find the forms of the teeth of the wheels which come into air-tight contact in the interior of the trough let us suppose the piston circles LM and NO, Fig. 47, to be described with the half-distance CD between the centers as a radius. From the point of contact F_2 or G_2 lay off the arcs $F_2F_1 = F_2F_3$ and $G_2G_1 = G_2G_3$, each equal to 30°; now generate the curves E_1G_1 and F_3G_3 by rolling the circle LM on NO. These arcs bound the float on the cross-arm attached to CA_1, and if in

like manner we roll the circle NO on ML, we obtain curves like F_1H_1, F_3H_3, which limit the floats on the cross-arms attached to the main arms DB_1 and DB_2 of the second wheel. While the shaft D and its arms DB_1, DB_2 turn with a right-hand rotation, and the shaft C with its arms turns with a left-hand rotation, the end F_1 of the curve F_1H_1 will slide on the curve E_1G_1, so that F_1 and G_1 will arrive at the same time at the line of

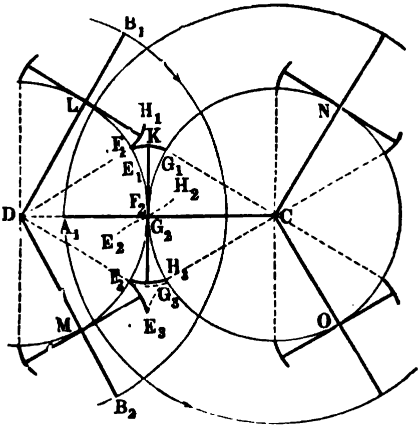

FIG. 47.

centers CD, there occupying the identical positions F_2 and G_2; but as the rotation is continued, the end G_2 of the curve $E_2G_2 = E_1G_1$ will slide on the arc F_3H_3, till the two curves occupy the positions E_3G_3 and E_3H_3. There the arms CA_1 and DB_1, etc., of the shafts C and D exchange their positions, a new contact beginning above, while the contact below (at G_3H_3) ceases, as may be seen in Fig. 48, where E_4G_4, F_4H_4 represent curves whose contact is beginning, and E_1G_1, F_1H_1 curves whose contact is ending.

The quantity of air which a float like DB_2, Fig. 46, takes out of M and discharges in the open air is equal to the sector $B_2DB_3 = B_1DB_2$ included between the two floats; on the other hand the volume of air forced back to M by one float, for example DB_1, is equal to the space $DF_1F_2F_3MD$, Fig. 47, which is bounded by the cross-arms LF_1, $F_1F_2F_3$, MF_3, and the portions DL and DM of the main arms. Therefore if we subtract the last space from the first, we shall obtain the actual discharge of air effected by

each float during a revolution. The space to be subtracted is also equal to the sectors DLF_2M increased by four small segments over the arc LF_2M, i.e., it is equal to the sector LF_2M + segment $F_1F_2G_1$ + segment $F_3F_2G_3$, and therefore the quantity discharged per float is V = sector B_1DB_2 − sector LDM − (seg. $F_1F_2G_1$ + seg. $F_3F_2G_3$) = annular ring LB_1B_2M minus twice segment $F_1F_2G_1$. If b is the width of a fan, r_1 the largest radius $DB_1 = DB_2$ of a float, r_2 the distance $DL = DM$ of a cross-arm from the axis D, and $\beta^\circ = B_1DC = CDB_2$, the half-angle between each pair of floats, then the volume of the annular space $B_1LF_2MB_2$ is

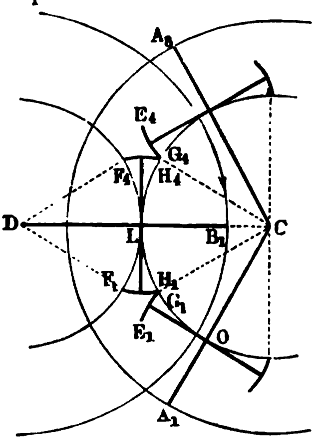

FIG. 48.

$$V_1 = \beta(r_1^2 - r_2^2)b,$$

and the volume of one of the small segments to be subtracted, for example that of the segment $F_1F_2G_1$ bounded by the circular arcs F_1F_2, G_1F_2, and the epicycloidal arc F_1KG_1, is found by the higher geometry to be

$$V_2 = 8\left(\frac{\beta}{2} - 2\sin\frac{1}{4}\beta\right)r_2^2 b;$$

consequently the delivery of a float is given by

$$V = V_1 - 2V_2 = \left(\beta(r_1^2 - r_2^2) - 16\left(\frac{\beta}{2} - 2\sin\frac{1}{4}\beta\right)r_2^2\right)b.$$

The whole delivery of the machine per revolution is $2 \times 3V = 6V$; now, if each of the shafts makes n revolutions per minute, the *theoretical discharge* per second will be

$$Q = \frac{n}{60}6V = \frac{nV}{10}.$$

If we substitute $\beta = \frac{\pi}{3}$, we obtain

$$V=\left(\frac{\pi}{3}r_1^2-\left(\frac{9}{3}\pi-32\sin 15^\circ\right)r_2^2\right)b$$

$$=[\pi r_1^2-(9\pi-24.8466)r_2^2]\frac{b}{3}=(\pi r_1^2-3.4277r_2^2)\frac{b}{3},$$

and therefore the whole volume of air discharged per second is

$$Q=(\pi r_1^2-3.4277r_2^2)\frac{nb}{30}.$$

In consequence of the clearance between the floats and the walls of the trough, etc., the actual discharge of air is only 70 per cent. of the volume given by this formula. In ordinary *Fabry* wheels $b=2$ m. [6.56 ft.], $r_1=1.7$ m. [5.58 ft.], and $r_2=1$ m. [3.28 ft.]; moreover, the R.P.M. are $n=36$ to 40, the difference of pressure between the outer air and that in the inlet is $h=4$ to 5 cm. [1.5 to 2 ins.] of water, and the efficiency $\eta=0.51$, when the steam-engine driving the wheels has theoretically a capacity for work of 15 horse-power.

Recently *Fabry* ventilators have been constructed in which each wheel has only two floats or wings, and the trough is built up on each side past the shaft centers. Further information concerning these ventilators is given in the second volume of *Pouson's* Traité de l'exploitation des mines de Houille, also by engineer *Jochams* in the Annales des travaux publics de Belgique, Tome XI and Tome XV.

The general arrangement and mode of action of a *rotary pump* which can also be employed to move air may be seen from the skeleton given in Fig. 49. *AEFG* is a cylindrical casing with two passages *M* and *N*, one of which acts as a suction- and the other as a delivery-pipe for the air; *AHBK* is a drum inclosed in this casing and placed eccentrically on the shaft *C*; finally, *ER* and *GS* are two pistons which can slide in the drum and are pressed outward by steel springs, the outer edges of the pistons pressing against the circumference of the casing. The casing is also touched by the drum at *A*, and in this way direct communication between the two passages is prevented. By means of the pistons *ER* and *GS* the rotating drum transfers from *M* to *N*, during each half-revolution, a volume of air equal to the space *HFK* between the drum and the casing.

Let r_2 represent the radius $DA=DB=DH$ of the drum, $r_1=CA=CE=CF$ represent the radius of the casing, d the eccen-

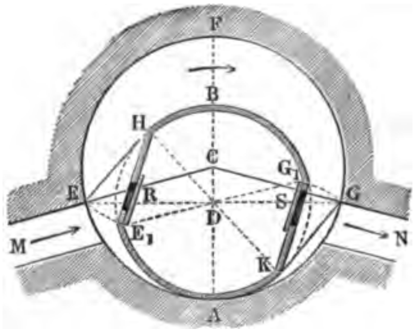

FIG. 49.

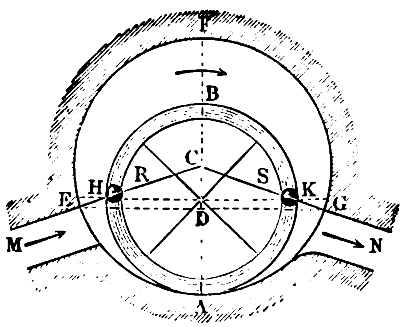

FIG. 50.

tricity CD of the axis D, and b the length of the casing and drum; then for the half-angle $ACH=ACK=\beta$ we have

$$\cos\beta=\frac{d}{r_1},$$

and for the area of the circular segment EFG

$$F_1=\pi r_1^2-(\beta r_1^2-dr_1\sin\beta);$$

also for the area of the semicircle HBK

$$F_2=\tfrac{1}{2}\pi r_2^2;$$

and therefore the volume of air which one piston transfers from M to N during a revolution is

$$V=(F_1-F_2)b=[\pi r_1^2-\beta r_1^2+dr_1\sin\beta-\tfrac{1}{2}\pi r_2^2]b.$$

Now, if the drum makes n R.P.M., the delivery per second is

$$Q=\frac{2nV}{60}=\frac{nV}{30}=[\pi r_1^2-\beta r_1^2+dr_1\sin\beta-\tfrac{1}{2}\pi r_2^2]\frac{nb}{30}.$$

Instead of employing the sliding pistons ER and GS, Fig. 49, we can use the turning pistons ER and GS, Fig. 50, provided the latter turn about the axis c of the casing $AEFG$, and pass out of the drum through ball-joints H and K. Then the quantity of air transferred by such a piston from M to N during each revolution is equal to the annular space $EFGKBH$=sector $EFGC$+ triangle HCK−semicircle HBK, all multiplied by the length of casing, and this gives

$$V=(\pi-\beta)r_1^2b+dr_2-\frac{\pi r_2^2b}{2}$$
$$=(\pi-\beta)r_1^2b+\left(\cot\beta-\frac{\pi b}{2}\right)r_2^2$$

where r_1 is again the radius CA of the casing, r_2 the radius DA of the drum, b the length of casing and drum, and β the half-angle ACH at the center for which we have

$$\cot\beta=\frac{d}{r_2},$$

d representing the eccentricity $CD=(r_1-r_2)$.

If the drum makes n R.P.M., the volume of air which this blower transfers from M to N per second is

$$Q=\frac{2Vn}{60}=\frac{Vn}{30}=\left[(\pi-\beta)r_1^2b+\left(\cot\beta-\frac{\pi b}{2}\right)r_2^2\right]\frac{n}{30}.$$

The *Mackenzie* blowers are constructed on this principle.*

* See Practical Mechanics Journal, Sept. 1857, and the Polytechnische Centralblatt, 1857.

In these blowers the diameter of the casing $2r_1$ is 1 m. [3.28 ft.], its breadth $b=0.9$ m. [2.95 ft.], and the diameter of the drum $2r_2=0.75$ m. [2.43 ft.]; consequently the eccentricity $d=r_1-r_2=0.125$ m. [4.92 ins.]. The floats are of sheet iron 6 to 12 mm. [$\frac{1}{4}$ to $\frac{1}{2}$ in.] thick, and their joints H and K are cylinders 75 mm. [3 ins.] in diameter and filled with soft metal; the number of floats is not two, but usually three or four. Leakage of air at A is prevented by wood covered with leather. In order that the axis C may keep a fixed position, the axis D is also fixed, the two being united by arms in the interior of the drum. The drum has two heads or bosses like an ordinary wagon-wheel and thus turns about D. These blowers are used in the United States principally to melt pig iron in cupolas, where they make 80 to 150 revolutions per minute, and deliver air having an effective pressure of $\frac{1}{4}$ to 1 kg. [0.5 to $2\frac{1}{2}$ lbs.].

Lemielle's ventilator has a similar arrangement, and this machine is outlined in Fig. 51. Here also a drum AB turns eccentrically in a cylindrical casing $AEFG$, but here the floats or pistons EH and GK are fastened to the circumference of

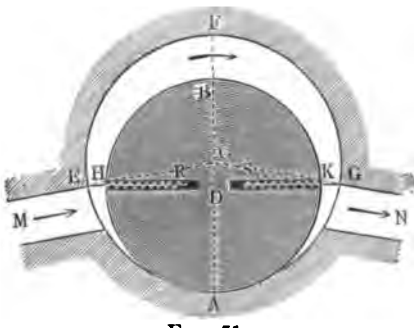

FIG. 51.

the drum by hinges H and K and are united by the joints E and G with the arms CE and CG, turning about the fixed axis C of the casing AF. In order that the floats may hug the drum when passing the point of contact A, the segments HE_1 and KG_1 are cut off from the drum, and in order that the arms CE and CG may pass air-tight through the walls E_1H and G_1K of the drum, the openings are covered with leather clacks R and S. In other respects the mechanism is like that of *Mackenzie's* blower.

The cross-section F of the air volume $EFGKG_1BHE$, which a float EH transfers from M to N during a revolution, is = circular segment $EFGE$ − triangle EHR + triangle GKS − half the cross-section $RHBG_1S$ of the drum, and as triangle EHR = triangle GKS, we have

$$F=\pi r_1^2-(\beta_1-\tfrac{1}{2}\sin 2\beta_1)r_1^2-\tfrac{1}{2}\pi r_2^2+(\beta_2-\sin\beta_2)\frac{r_2^2}{2}$$
$$=(\pi-\beta_1+\tfrac{1}{2}\sin 2\beta_1)r_1^2-(\pi-\beta_2+\sin\beta_2)\frac{r_2^2}{2},$$

where r_1 = radius $CA=CE$ of the casing, r_2 the radius $DA=DH$ of the drum, β_1 the central angle ECA, and β_2 the central angle $E_1DH=G_1DK$. When the eccentricity $CD=d$ we have

$$\cos\beta_1=\frac{d}{r_1},$$

and with a length of float $EH=s$

$$\sin\frac{\beta_2}{2}=\frac{s}{2r_2}.$$

If b is the breadth of the wheel space measured parallel to the axis, and n the number of revolutions of the wheel per minute, the *theoretical volume* of air delivered per second is

$$Q=\frac{2Fbn}{60}=\frac{Fbn}{30}$$
$$=\left((\pi-\beta_1+\tfrac{1}{2}\sin 2\beta_1)r_1^2-(\pi-\beta_2+\sin\beta_2)\frac{r_2^2}{2}\right)\frac{bn}{30}.$$

Such ventilators have also been constructed with three or more floats. The theoretical delivery is not essentially changed by the use of more than two floats, but the loss of air due to the clearance is somewhat reduced.

According to experiments made, the actual delivery of *Lemielle's* ventilators with two floats is given by

$$Q_1=Q-0.39\sqrt{h}\times\frac{n}{60}=(2Fb-0.39\sqrt{h})\frac{n}{60},$$
$$\left[Q_1=Q-68.8\sqrt{h}\frac{n}{60}=(2Fb-68.8\sqrt{h})\frac{n}{60}\right],$$

where the height of the manometer or the depression of the air pressure h at the suction-orifice is expressed in millimeters [inches] of water, and the air volume Q in cubic meters [cu. ft.].

This formula, of course, holds only for a ventilator of this construction and size. In the ventilator on which this formula was based $2r_1=3.95$ m. [12.96 ft.] and $2r_2=3$ m. [9.84 ft.]; consequently $d=r_1-r_2=0.475$ m. [1.56 ft.], $b=2.1$ m. [6.89 ft.], $n=20$ to 30, and the height of the water-manometer $h=12$ to 36 mm. [0.5 to 1.5 m.].

The details of a Lemielle ventilator are shown by the two sections given in Fig. 52 and Fig. 53. The casing $AEFG$ is of

FIG. 52.

wood or iron, or it may be of masonry lined with cement. The drum $AHBK$ has an iron skeleton and a wooden rim, and turns about the journals B and D_1 of a bent axle CD, which is fixed in the bearing L. The wooden floats EH and GK are fastened to the drum by the hinges H and K, and are connected by the arms CE and CG with the shaft C. The upper hub V of the drum inclosing the pin D_1 and in turn inclosed by the bearing W fixed to the beams U carries the boss X of a crank Y which is driven by the connecting-rod of a steam-engine. The horizontal cylinder Z of this engine likewise rests on the long beams U.

Root's blower also comes under the head of rotary blowers; mention was made of it in the Mechanics of Pumping Ma-

chinery when rotary pumps were discussed. Recently this simple arrangement has come into extensive use in foundries

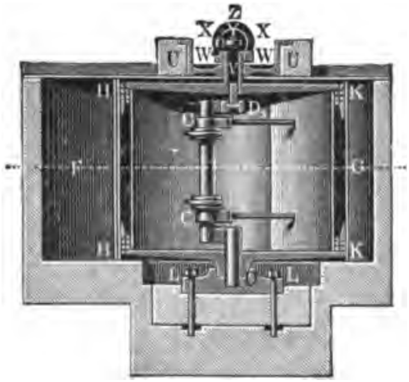

FIG. 53.

and blacksmith-shops. It is shown in Fig. 54 and Fig. 55, and consists principally of two rotating bodies B and B' which, fastened to parallel shafts CC', rotate in opposite directions

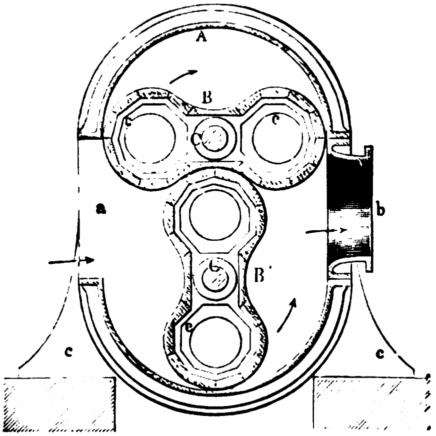

FIG. 54.

with equal velocity. As the form of these bodies is such that they touch each other like toothed wheels, and as they are inclosed in a casing A fitting them as closely as practicable, it is easy to see how the air between the piston and casing is forced through the opening b into the air-pipe, while fresh air continually enters the casing through the suction-opening a. We

also see how *each* shaft forces the air out of the space V twice during every revolution, so that the theoretical discharge per minute is $Q=4nV$, where n represents the R.P.M. of the machine, i.e., of each of the two shafts.

These blowers are usually run at a speed of 250 to 300 R.P.M., in consequence of which velocities they are subjected to considerable vibrations and work with a great deal of noise. To diminish this defect and prevent leakage the two rotating pistons were formerly made of wood as the figures show, but recently cast iron has been employed and the two shafts placed

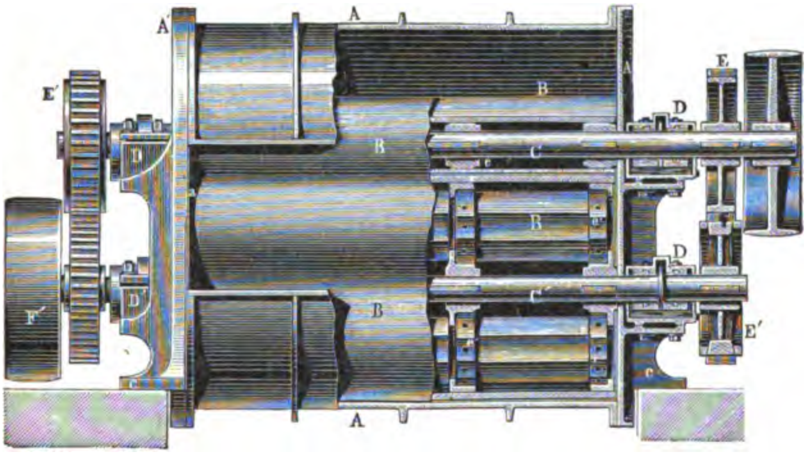

FIG. 55.

side by side instead of one over the other so that the suction-opening comes below and the delivery-opening above. The shafts are driven by belt-pulleys F, and the toothed wheels E cause the shafts to assume the proper relative motion.

As the two pistons do not actually touch the casing there will always be a certain space between them, and of course this will make the actual delivery smaller than the theoretical, the ratio of the two becoming smaller the greater the pressure of the blast generated. The efficiency also diminishes rapidly with the increase of blast pressure. According to *Hartig's* experiments * the coefficient of discharge w and the efficiency η

* Versuche ueber der Leitung und Arbeitsverbrauch von Werkzeugmaschinen, Leipzig, 1873.

for a pressure h in millimeters [inches] of water was

$$h=138 \text{ mm. [1.5 ins.]} \ldots\ldots\ldots\ldots w=0.79; \quad \eta=0.405;$$
$$h=820 \text{ mm. [32.28 ins.]} \ldots\ldots\ldots w=0.12; \quad \eta=0.143$$

According to *Ledebur*,* *Root's* blowers can be recommended for pressures up to 400 mm. [16 ins.] of water, and we may assume on an average a coefficient of discharge of $w=0.75$ and $\eta=0.45$ for them.

§ 25. **Fans.** — In the piston-blowers discussed above, the air is directly compressed by the surface of a rigid body; but in the ventilators or *fan-blowers* now to be discussed the pressure of the air is changed by a change in its state of motion; nevertheless in both kinds of blowers the motion of the air results from the change of pressure. A change in the state of motion of a body consists either of a change of the *direction* of the motion, or in the velocity of the motion, or in both at once.

In the so-called *centrifugal* blowers or *centrifugal ventilators* it is principally the change in the direction of motion or the centrifugal force resulting from it which changes the pressure of the air; in the *screw ventilators* and in the *pipe ventilators*, which resemble reaction turbines, the change of pressure of the air is principally effected by a change of velocity. All these ventilators or fans can be employed as exhausters as well as blowers.

Centrifugal fans consist principally of a simple wheel provided with vanes and surrounded by a casing; when a fan turns, the centrifugal force causes air to be drawn in through an orifice near its axis, and this air is driven out again through an orifice in the circumference of the casing. According as the first or second orifice communicates with a closed space the fan will act, relatively to this space, as an *exhauster* or *blower*. In the first case the air that has been sucked in is forced out into the atmosphere through the orifice in the circumference of the casing, and in the second case the atmospheric air is sucked in through the opening near the shaft. As for the rest,

* Die Verarbeitung der Metalle auf Wege von A. *Ledebur*, Braunschweig, 1877.

the action of the fan is the same in both cases, and the difference between them is that the pressure in the inclosed space is smaller in one case and larger in the other than the pressure of the atmosphere.

The vertical section of a centrifugal fan is shown in Fig. 56. *BCB* is the fan on the shaft *C*, and *AB* its vanes; moreover, *ACA* is the inlet at the back of the casing, and *EF* the discharge-pipe with the outlet *F*. The vanes are either flat or curved

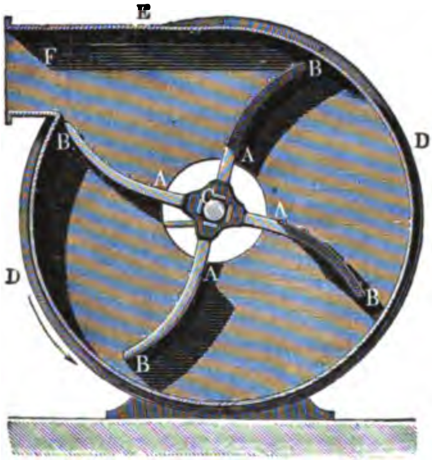

FIG. 56.

and in the first place are set radially or obliquely to the radius; they are either rectangular or trapezoidal. Ordinarily only four to eight vanes are employed.

Screw ventilators have vanes which are set obliquely to the plane of rotation and therefore are not essentially different from the wheels of ordinary windmills (see Vol. II), nor from the wheels of screw propellers (see Vol. III, 2, chap. 3). These ventilators, too, have ordinarily from three to eight vanes.

With a greater number and extent of vanes the spaces between them become passages and pipes, and the ventilators become *pipe* or *reaction ventilators*. As the action of screw ventilators is to be judged according to the rules for the wheels of windmills and screw propellers, so the reaction ventilators are closely related to the reaction turbines. The difference of pressure ordinarily generated by fans is a very small part,

say 1 to 5 per cent., of the original pressure of the air, and the change of density of the air while flowing through fans is small. *Hence* we may safely treat the air as if it were water, and may discuss the action of fans, particularly of pipe ventilators, as we should the action of centrifugal pumps and turbines.

In each fan we should notice the inlet and outlet connected with the casing. The object of the inlet is to lead the air to be moved from the suction-space to the inner periphery of the wheel with as little loss as possible. In suction-fans the inlet is connected by a pipe or passage with the space from which the air is to be removed, as in mines where ventilation is effected by a fan; in fan-blowers the air is usually taken directly from the atmosphere. The outlet, on the other hand, serves to lead the air, which is leaving the outer circumference of the wheel with great velocity, in a suitable manner to the conduit which is to carry it off. But in suction-fans the forced-out air usually passes directly into the atmosphere, and it is only when the air contains hurtful ingredients (poisonous gases, particles of metal, emery dust) that it is discharged into a special conduit, in which case the fan exerts at the same time a sucking and a forcing action. Generally speaking, there is no well-defined difference between suction-fans and fan-blowers, for every fan may exert a sucking as well as a blowing action.

Sometimes fans are distinguished as open or closed fans according as the air enters freely into the atmosphere at the whole periphery of the outlet or is led through a gradually widening space to the air-pipe. This outlet must gradually increase in order that the great velocity with which the air leaves the wheel may be diminished as much as possible, thus reducing the loss due to the living force of the escaping air. In closed fans the aforesaid diminution of velocity causes an increase of pressure in the air-conduit; in open fans a ring-shaped outlet may be formed by simply widening the sides of the casing in all directions,* which likewise provides a gradual increase of cross-section for the escaping air and thus accomplishes the required diminution of velocity.

§ 26. **Velocity of Fans.**—The law expressing the relation between the velocity of the centrifugal fan and the pressure of

* See *Rittinger*, Centrifugalventilatoren.

the inclosed air can be most easily seen if we first assume that the vanes are flat, radial surfaces, and that the velocity of efflux is very small in comparison with the velocity of rotation at the circumference. Then the velocity of the air in a

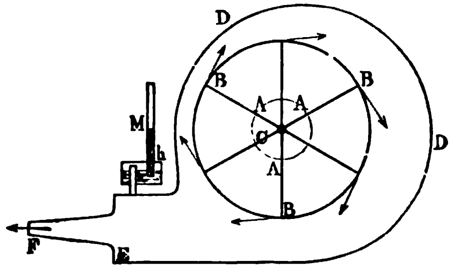

FIG. 57.

radial direction is also small and may be neglected in judging of the performance of the wheel.

If the fan ABC, Fig. 57, turns with the angular velocity ω, the velocity of a point at a distance x from the axis is

$$u = x\omega;$$

and if at this distance there is an element of air having a cross-section unity, a thickness dx, and the specific weight γ, the centrifugal force of this element will be

$$dp = \omega^2 x \frac{\gamma dx}{g}. \quad \text{(See Vol. I, sec. v, chap. 3.)}$$

Now, according to *Mariotte's law*,

$$\gamma = \frac{0.000125p}{1+0.00367t} = \phi p \text{ kg.} \left[\gamma = \frac{0.0055p}{1+0.00203(t-32)} \text{ lbs.}\right],$$

where p is the pressure of the air in kilograms per square meter [pounds per square inch], and ϕ is a coefficient depending on the temperature t, which is usually constant. Consequently

$$\frac{dp}{p} = \frac{\phi}{g}\omega^2 x dx, \quad \log_e p = \frac{\phi}{g}\frac{\omega^2 x^2}{2}.$$

Now, if r_1 represents the inner and r_2 the outer radius of the wheel, and if p_1 and p_2 represent the pressure of the air at the

inner and outer circumferences respectively, we obtain by integration

$$\int_{p_1}^{p_2} \frac{dp}{p} = \frac{\phi}{g} \omega^2 \int_{r_1}^{r_2} x dx,$$

i.e.,

$$\log_e \frac{p_2}{p_1} = \frac{\phi}{g} \omega_2 \frac{r_2^2 - r_1^2}{2} = \frac{\phi}{g} \frac{u_2^2 - u_1^2}{2},$$

or

$$p_2 = p_1 e^{\frac{\phi}{g}\left(\frac{u_2^2 - u_1^2}{2}\right)},$$

where u_1 and u_2 represent the inner and outer velocities of the wheel and where e is the base of the natural system of logarithms.

As the exponent

$$\frac{\phi}{2g}(u_2^2 - u_1^2) = \frac{0.000125}{2 \times 9.81}(u_2^2 - u_1^2) = 0.0000064(u_2^2 - u_1^2)$$

is always very small, it is sufficiently accurate to place

$$e^{\frac{\phi}{2g}(u_2^2 - u_1^2)} = 1 + \phi \frac{u_2^2 - u_1^2}{2g} = \frac{p_2}{p_1},$$

and we then get

$$p_2 - p_1 = \phi p_1 \frac{u_2^2 - u_1^2}{2g} = \gamma_1 \frac{u_2^2 - u_1^2}{2g}.$$

If, instead of the pressures p_1 and p_2, we use the corresponding heights b_1 and b_2 of the water-barometer, we obtain

$$b_2 - b_1 = \phi b_1 \frac{u_2^2 - u_1^2}{2g} = \frac{\gamma_1}{\gamma_0} \frac{u_2^2 - u_1^2}{2g} = \frac{1}{\epsilon_1} \frac{u_2^2 - u_1^2}{2g},$$

where $\epsilon_1 = \frac{\gamma_0}{\gamma_1}$ is the ratio of the density of the water to that of the air entering at A. For the case in which the outer radius r_2 is considerably greater than the inner radius r_1 we can write approximately $b_2 - b_1 = \frac{u_2^2}{2g\epsilon_1}$.

In the case of an exhauster, if the air at the outer circumference of the wheel flows directly into the atmosphere, its

velocity of efflux, which is there nearly equal to the circumferential velocity, becomes zero without exerting any action. In this case b_2 equals the barometer height, b_0, of the atmosphere; hence we have a negative manometer height in the supply-passage which is equal to

$$h=b_0-b_1=\frac{u_2^2}{2g\varepsilon_1}.$$

In the corresponding case of a blower without diffuser we have

$$h=b_2-b_1=b_2-b_0=\frac{u_2^2}{2g\varepsilon_1}.$$

On the other hand, if we provide the fan with *flaring sides* * or surround it with a diffuser (see Vol. III) as shown in Fig. 57, whereby the velocity u_2 of the air leaving the wheel is gradually reduced almost to zero, the living force will be transformed into pressure measured by a water column of a height $\frac{u_2^2}{2g\varepsilon_1}$, and this will give us for the outer pressure of the air, which we will designate by b_2', the relation, for an exhauster,

$$b_0=b_2'=b_2+\frac{u_2^2}{2g\varepsilon_1}=\left(b_1+\frac{u_2^2}{2g\varepsilon_1}\right)+\frac{u_2^2}{2g\varepsilon_1}=b_1+2\frac{u_2^2}{2g\varepsilon_1}.$$

Calling the manometer height H, we have

$$H=b_2'-b_1=b_0-b_1=2\frac{u_2^2}{2g\varepsilon_1}=2h,$$

and for a blower

$$b_2'=b_2+\frac{u_2^2}{2g\varepsilon_1}=\left(b_1+\frac{u_2^2}{2g\varepsilon_1}\right)+\frac{u_2^2}{2g\varepsilon_1}=b_0+2\frac{u_2^2}{2g\varepsilon_1},$$

from which

$$H=b_2'-b_0=2\frac{u_2^2}{2g\varepsilon_1}=2h,$$

* See *Rittinger*, Centrifugalventilatoren.

which is double the manometer height for a fan from which the air flows out freely from all points of the circumference.

In fan-blowers the air flowing out from the circumference of the wheel is taken up by the casing which leads it to the blast-pipe, so that this casing, like the diffuser of a suction-fan, reduces the velocity of efflux of the air nearly to zero. Therefore the barometer height b_2 at the beginning of the blast-pipe is

$$b_2 = b_1 + 2\frac{u_2^2}{2g\varepsilon_1} = b_0 + 2\frac{u_2^2}{2g\varepsilon_1},$$

for b_1 is here equal to the barometer reading b_0 of the outer air, with which the inlet is in direct communication. Hence the manometer height at the beginning of the blast-pipe is likewise

$$h = b_2 - b_0 = 2\frac{u_2^2}{2g\varepsilon_1} = \frac{u_2^2}{g\varepsilon_1}.$$

Owing to the friction of the air at the sides of the casing and other disturbing influences, the manometer height in both ventilators becomes considerably smaller than $\frac{u_2^2}{g\varepsilon_1}$ in cases where the radial velocity of the air in the wheel is not very small in comparison with its circumferential velocity.

The work L theoretically required is

$$L_0 = Qh\gamma_0 = Q\frac{u_2^2}{g\varepsilon_1}\gamma_0 = Q\frac{u_2^2}{g}\gamma_1 \qquad \left(\varepsilon_1 = \frac{\gamma_0}{\gamma_1}\right),$$

where Q is the quantity of air delivered per second. The work actually required is considerably greater and equal to

$$L = \frac{Qh\gamma_0}{\eta},$$

where η is the efficiency of the fan, which experience shows must not be taken greater than 0.3.

In most cases, however, the relative velocities of the air within the wheel are not so small that they can be neglected; moreover, the vanes are usually curved, so that the action of this class of fans needs a special investigation.

For this purpose let r_1 again represent the inner and r_2 the outer radius of the wheel ABC, Fig. 58, which we will suppose to have a number of curved vanes of the form AB. We will assume that the air entering the wheel in the direction of the axis C is uniformly and gradually deflected in all directions by the conoidal inlet, so that any particle of air will enter the

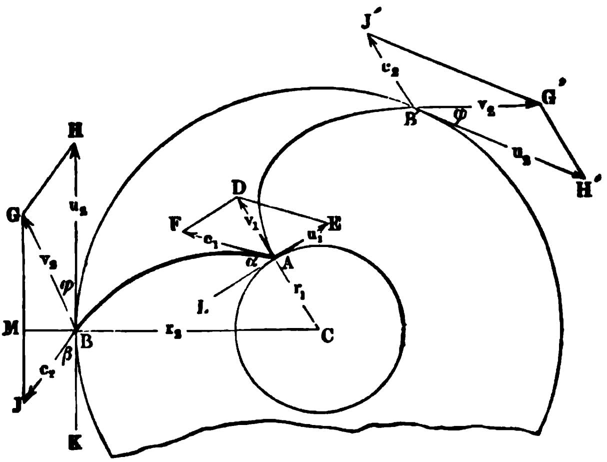

FIG. 58.

wheel at A with the radial velocity $AD=v_1$. In order that the air may enter the rotating wheel without impact we must have for the interior circumferential velocity $u_1=AE$ of the wheel the relation (see Fig. 58) $u_1=v_1 \cot \alpha$, where α is the inner angle FAL made by the vane.

Under this condition of freedom from impact we have for the relative velocity $c_1=AF$, with which the particle of air begins its travel along the vane, the equation

$$c_1^2=v_1^2+u_1^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The rotating vane now continually exerts an accelerating action on the particle of air till it leaves the wheel at the outer circumference. Let us suppose that this occurs at the instant at which the end B of the vane has reached B'; then the particle of air has described an *absolute* path which is represented, say, by the curve AB', this curve being tangential to the radial velocity $v_1=AD$ at the entrance A. At B' the air enters the

outlet with an absolute velocity $B'G'=v_2$ which is tangential to its absolute path AB', and must be regarded as the resultant of two other velocities, namely, the outer circumferential velocity $u_2=B'H'=BH$ of the wheel, and the relative velocity $c_2=B'J'=BJ$ with which the air moves along the last element of the vane. Therefore if ϕ represents the angle $G'B'H'=GBH$ at which the air leaves the circumference of the wheel, we have

$$c_2^2=v_2^2+u_2^2-2v_2u_2\cos\phi. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The entrance velocity $v_1=AD$ of the air is, as in all suction, generated by the pressure of the atmosphere.

If we represent the specific weight of water, 1000 kg. [62.5 lbs. per cu. ft.], by γ_0, and that of the air entering at A by γ_1, and the coefficient of resistance to entrance into the wheel by ζ_1, then to generate the velocity v_1 of the air there will be needed a head of water expressed by

$$(1+\zeta_1)\frac{v_1^2}{2g}\frac{\gamma_1}{\gamma_0}=(1+\zeta_1)\frac{v_1^2}{2g\epsilon_1},$$

where ϵ_1 is the ratio $\frac{\gamma_0}{\gamma_1}$ of the specific weights of the water and the air. Consequently the pressure of the air at A is less than that in the supply-pipe just outside the inlet by an amount measured by the head consumed in generating this velocity. The aforesaid pressure in the supply-pipe for a fan-blower, which takes the air directly from the atmosphere, is equal to the height b of the water-barometer. On the other hand, for a suction-fan it is smaller by a certain amount, corresponding to the resistances in the suction space. Let us designate by h_1 the height of the water-manometer on the suction-pipe just before the entrance to the fan: h_1 having a negative value for suction-fans, and being equal to zero for fan-blowers; the head x measuring the pressure of the air at the entrance A is

$$x=b+h_1-z_1-\frac{v_1^2}{2g\epsilon_1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where $z_1=\zeta_1\frac{v_1^2}{2g\epsilon_1}$ is the head of water corresponding to the resistance to entrance.

Also let h_2 be the height of the water-manometer for the air at the end of the outlet, i.e., at the place where the air-pipe is attached to the fan-blowers or where in suction-fans the air flows into the atmosphere. Moreover, let w be the velocity with which the air leaves the outlet at the place either to enter the air-pipe or the atmosphere, as the case may be, and let z_2 be the head measuring the resistance in the outlet. The air which leaves the wheel at B with the absolute velocity v_2 and also possesses a certain pressure, measured by the head of water y, must not only be able to overcome the pressure $b+h_2$ at the orifice of the outlet, and the resistance z_2, but also to impart a velocity w to the air. Hence if γ_2 represents the specific weight of the air where it leaves the wheel, and if we place $\frac{\gamma_0}{\gamma_2}=\varepsilon_2$, we have the equation

$$y+\frac{v_2^2}{2g\varepsilon_2}=b+h_2+z_2+\frac{w^2}{2g\varepsilon_2}. \quad . \quad . \quad . \quad . \quad . \quad (4)$$

We must here remark that in fan-blowers h_2 is a positive quantity which, under some circumstances, may become nearly equal to 1 m. [3.28 ft.]. On the other hand, in open fans we must place $h_2=0$, for here the air is forced directly into the atmosphere. In this case w represents the velocity of the air at the place where it enters the atmosphere, i.e., where the diffuser stops if it has been used. With a small difference in the pressures at the inlet and outlet it is sufficiently accurate to place the specific weights γ_1 and γ_2 equal to each other, and then $\varepsilon_1=\varepsilon_2$, and we may write them more simply as ε.

In addition to the four equations obtained for the entrance of the air to the wheel and its exit, we now obtain a fifth, which expresses the accelerating influence of the rotating vanes. From what was said in Vol. I, sec. v, chap. 3, we know that the increase of living force, experienced by the air in passing through the fan from A to B', is proportional to the heads of the wheel velocities at A and B; that is, we have

$$y+\frac{c_2^2}{2g\varepsilon}-\left(x+\frac{c_1^2}{2g\varepsilon}\right)+z_r=\frac{u_2^2-u_1^2}{2g\varepsilon}, \quad . \quad . \quad . \quad (5)$$

where z_r is the head representing the resistance experienced during the passage through the wheel. If we substitute in this

equation (5) the values of c_1, c_2, x, and y, given by equations (1) to (4), we get, after reduction,

$$h_2-h_1+z_1+z_2+z_r+\frac{w^2}{2g\varepsilon}=\frac{2v_2u_2\cos\phi}{2g\varepsilon}.$$

If we place $(h_2-h_1)=h$ and $z_1+z_2+z_r=z$, we also get

$$g\varepsilon(h+z)+\frac{w^2}{2}=v_2u_2\cos\phi. \quad . \quad . \quad . \quad . \quad . \quad (6)$$

According to the figure we have

$$v_2=u_2\frac{\sin\beta}{\sin(\beta+\phi)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

and substituting this we obtain

$$u_2=\sqrt{\left(g\varepsilon(h+z)+\frac{w^2}{2}\right)\frac{\sin(\beta+\phi)}{\sin\beta\cos\phi}}, \quad . \quad . \quad . \quad . \quad (8)$$

which equation gives the circumferential velocity u_2 when the difference of pressure h, the angle β, and the angle of exit ϕ are known. When the vanes end radially, i.e., $\beta=90°$, this equation becomes

$$u_2=\sqrt{g\varepsilon(h+z)+\frac{w^2}{2}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (8^a)$$

Equation (6) gives us the absolute velocity of efflux v_2, and from this we find the radial velocity with which the air leaves the wheel to be

$$BM=v_2\sin\phi.$$

Now if l_1 and l_2 represent the depth in the clear at the inner and outer circumference of the wheel, we have for the quantity of air Q the relation

$$Q=2\pi r_2l_2v_2\sin\phi=2\pi r_1l_1v_1, \quad . \quad . \quad . \quad . \quad . \quad (9)$$

from which follows

$$v_1=\frac{r_2l_2}{r_1l_1}v_2\sin\phi. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

We then obtain the angle α made by the vane with the inner circumference, namely,

$$\tan\alpha=\frac{v_1}{u_1}, \text{ etc.}$$

On the other hand, if the exit angle ϕ is not known beforehand, but some other quantity is given, say the entrance velocity v_1 of the air, we obtain an equation for determining the circumferential velocity as follows: In equation (6) $v_2\cos\phi$ evidently represents the tangential component MG of the velocity of efflux. Now, according to the figure we have

$$MG=BH-MJ=u_2-v_2\sin\phi\cot\beta,$$

and combining this with (10), we have

$$v_2\cos\phi=u_2-\frac{r_1l_1}{r_2l_2}v_1\cot\beta,$$

so that now (6) becomes

$$g\epsilon(h+z)+\frac{w^2}{2}=u_2\left(u_2-\frac{r_1l_1}{r_2l_2}v_1\cot\beta\right).$$

Solving this quadratic equation, we have

$$u_2=\frac{r_1l_1}{2r_2l_2}v_1\cot\beta+\sqrt{\left(\frac{r_1l_1}{2r_2l_2}v_1\cot\beta\right)^2+g\epsilon(h+z)+\frac{w^2}{2}}. \quad . \quad (11)$$

For $\beta=90°$, i.e., for the case of radial vanes, the equation reduces to equation (8^a).

From the circumferential velocity u_2 we obtain the number of revolutions n per minute according to

$$60u_2=2\pi r_2n, \text{ etc.}$$

From equation (11) we see that for a given difference of pressure h the circumferential velocity of the wheel becomes less the smaller $\cot\beta$, other things being equal. Therefore, if we make $\beta=90°$, i.e., let the vanes end radially, we have $\cot\beta=0$, and obtain for this case

$$u_2=\sqrt{g\epsilon(h+z)+\frac{w^2}{2}}. \quad . \quad . \quad . \quad . \quad . \quad (12)$$

For this reason *Rittinger* recommends the use of vanes that end radially, for in vanes that curve back, i.e., those in which $\beta < 90°$, the circumferential velocity needed to produce a certain difference of pressure h is greater, which involves increased friction at the journals and wear of the bearings.

We will now mention the various relations and dimensions of fans; usually the difference h of the pressures is small. In suction-fans we may assume this difference to be 0.03 to 0.06 m. [1.18 to 2.36 ins.], while in fan-blowers it is usually between 0.15 and 0.5 m. [5.9 to 19.68 ins.].

According to *Rittinger* it sometimes becomes as large as 0.8 m. [31.5 ins.]. Equations (8) and (12) show that the velocity of the wheel varies with h.

The ratio of the radii of the wheel in suction-fans is usually $\frac{r_1}{r_2}=\frac{1}{2}$, and in fan-blowers $\frac{r_1}{r_2}=\frac{1}{3}$. For the velocity v_0 of the air entering the inlet *Rittinger* gives 10 m. [32.8 ft.] as a suitable value, so that for a delivery of Q cubic meters [cu. ft.] we have for the radius r_0 of the inlet orifice

$$r_0=\sqrt{\frac{Q}{\pi v_0}}=0.18\sqrt{Q}\text{ m.}\quad [r_0=0.985\sqrt{Q}\text{ ft.}].$$

It is well to take the inner radius r_1 of the wheel equal to the radius r_0 of the inlet orifice, or, taking the thickness of the vanes into account, it may, according to *Fink*, be taken equal to about $1.2r_0$; if we make the entrance velocity v_1 of the air into the wheel equal to the velocity v_0 at the inlet to the casing, we can find the axial width l_1 of the vanes from

$$Q=2\pi r_1 l_1 v_1=\pi r_0^2 v_0,$$

which for $r_1=r_0$. gives $l_1=0.5r_0$,

and for $r_1=1.2r_0$ gives $l_1=0.42r_0=0.35r_1$.

This width applies to fans which receive air at only one side, and when air is received at both sides we must assume half the delivery for each side; with this as a basis, we can determine the radius r_0 of the entrance orifice, and the width of the vanes on each side of the middle plane will be equal to

the above values; the total width will therefore be double those already found. Often the width l_2 of the fan at the outer circumference is equal to that at the inner, but frequently the width of the vane is diminished toward the outside, the sides of the casing converging. Sometimes the number of revolutions of the wheel is as great as 2000 per minute.

The work per revolution can be placed equal to

$$L = Q\gamma_0\left(h + z + \frac{w^2}{2g\varepsilon}\right), \quad \ldots \ldots \quad (13)$$

so that, neglecting the friction of the journals, the efficiency becomes

$$\eta = \frac{h}{h + z + \dfrac{w^2}{2g\varepsilon}}. \quad \ldots \ldots \quad (14)$$

To determine the value of z it is well to use the dynamometric tests made on fans. According to these tests the efficiency of fans may be assumed to average about 30 per cent. Thus *Hartig's* experiments give for ordinary conditions an efficiency between 0.24 and 0.36. The extensive experiments made by *Rittinger* with specially constructed fans gave in the most favorable case the efficiency 0.28 for a suction-fan and 0.30 for a fan-blower. This most favorable value occurred with a certain circumferential velocity which in the fan-blowers was 1.77 times as great as the theoretical value obtained from $u_2 = \sqrt{g\varepsilon h}$, which formula may be obtained from (8^a) when the values of z and w are neglected. These two experimental results agree very well with the theory given above, for if we place

$$u_2 = \sqrt{g\varepsilon(h+z) + \frac{w^2}{2}} = 1.77\sqrt{g\varepsilon h},$$

we obtain a value for the efficiency

$$\eta = \frac{h}{h + z + \dfrac{w^2}{2g\varepsilon}} = \frac{1}{1.77^2} = 0.32.$$

If we compare this result with the experimentally determined efficiency 0.28, which takes into account the journal

friction, we see that the two values agree very fairly. *Rittinger* himself, after subtracting the loss due to journal friction, finds the efficiency of the fan to be 0.31, very nearly equal to that given by equation (8^a) when the actually observed velocity u_2 is substituted in it.

§ 27. The Designing of Fans.—In order that the air may enter the wheel without contractions or the formation of eddies, a conoidally shaped inlet is employed, which gradually changes the axial direction of the approaching air into a radial direction. If we assume, as stated above, that the air enters the wheel with the same velocity v_0 which it possesses in the suction-pipe, we can construct the inlet in the following manner: If $AB=r_0$, Fig. 59, is the radius of the suction-pipe and $FG=r_1$ is the inner radius of the wheel whose width $FE=l_1$ in the clear was determined so that $2\pi r_1 l_1 = r_0^2\pi$, we can first draw a quadrant CD which bisects the width FE of the vane at C and bisects the radius AB at D. If we regard this arc as the path of the average particle of air, then the problem is to so determine the boundaries ALE and BKF of the inlet that the air will everywhere find a passage area πr_0^2. If we draw through any point J of the arc the radius OK, the passage area corresponding to the point J is given by the frustum of a cone having BG as an axis and the distance LK as a generatrix. If we place the length $LJ=JK=x$, and represent the distance JH of the point J from the axis by ρ, then we must place the aforesaid conical surface $2\pi\rho 2x$ equal to πr_0^2. Now the figure shows that

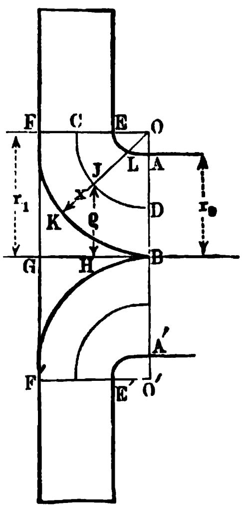

FIG. 59.

$$\rho = JH = r_1 - \left(r_1 - \frac{r_0}{2}\right)\cos\psi,$$

the angle DOJ at the center being represented by ψ.

We therefore have

$$2\pi\left(r_1 - \left(r_1 - \frac{r_0}{2}\right)\cos\psi\right)2x = \pi r_0^2,$$

from which follows

$$x=\frac{1}{4}\frac{r_0^2}{r_1-\left(r_1-\frac{r_0}{2}\right)\cos\psi}.$$

If we substitute in this $r_1=vr_0$, the expression becomes

$$x=\frac{1}{4}\frac{r_0}{\nu-(\nu-\frac{1}{2})\cos\psi};$$

for example, for $\nu=1.2$ we get

$$x=\frac{r_0}{4.8-2.8\cos\psi}.$$

From this we can obtain, for assumed values of ψ between 0° and 90°, the corresponding values of x, and then draw the two meridian lines ALE and BKF for the conoidal inlet.

As regards the form of the fan vane we may first remark that its inner angle α depends on

$$\tan\alpha=\frac{v_1}{u_1},$$

which is the condition that there shall be no impact at the entrance. This condition cannot be satisfied when vanes are employed that begin radially, i.e., when $\alpha=90°$, for in this case the entrance velocity v_1 would have to be infinitely great. Therefore when radial vanes are employed the entrance of the air is attended with a loss of work $\frac{u_1^2}{2g}Q\gamma$. The loss due to impact is generally expressed by

$$\frac{(u_1-v_1\cot\alpha)^2}{2g}Q\gamma,$$

for the tangential component of the entrance velocity $v_1\cot\alpha$ is suddenly transformed into u_1. *Rittinger's* experiments also showed that the performance was small when flat radial vanes were employed, the highest efficiency in this case being only 0.08.

If we give the vanes a flat form AB', Fig. 60, which cuts the inner circumference of the wheel at an angle α, the triangle

ABC gives for the angle β, formed by the vane with the outer circumference, the expression $\frac{r_1}{r_2}=\frac{\cos\beta}{\cos\alpha}$.

With this form of vane the ratio of the relative velocities

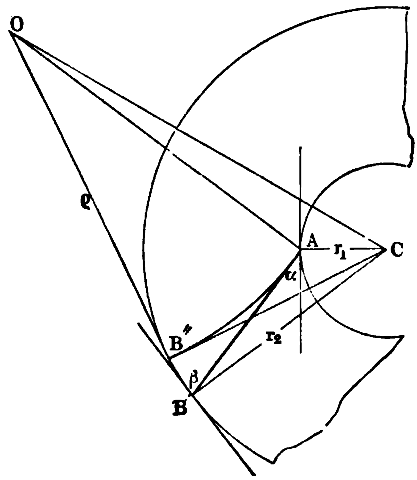

FIG. 60.

of the air at the entrance and exit becomes, neglecting the thickness of the vanes,

$$\frac{c_2}{c_1}=\frac{r_1l_1\sin\alpha}{r_2l_2\sin\beta}=\frac{l_1\tan\alpha}{l_2\tan\beta}.$$

In order that the efflux may take place with full buckets we must have $c_2 \leqq c_1$; hence we must make $l_2\tan\beta \geqq l_1\tan\alpha$. But with curved arms for which the aforesaid dependence of β or α does not exist we have

$$\frac{c_2}{c_1}=\frac{r_1l_1\sin\alpha}{r_2l_2\sin\beta}.$$

The remarks made in connection with the form of the blades of centrifugal pumps apply here also. *Rittinger* recommends that circular arcs be chosen and that the outer vane-angle β be assumed equal to 90°. The center O of such a vane AB'' lies on a perpendicular to the direction AB' of the vane at A, and

the radius $\rho = OB''$ is obtained by equating the two values given for CO by the triangle $CB''O$ and CAO. Accordingly we have

$$\overline{CO}^2 = r_2^2 + \rho^2 = r_1^2 + \rho^2 + 2r_1\rho \cos \alpha,$$

from which follows

$$\rho = \frac{r_2^2 - r_1^2}{2r_1 \cos \alpha}.$$

If we do not make the outer angle $\beta = 90°$, but choose some other value for β, the arc for the vane can easily be constructed as follows:

In the first place the center O, Fig. 61, of the arc constituting the vane lies on the straight line AO, which is perpendicular to the direction AE of the vane at its initial point. Now, if we draw through A the straight line AD, making the angle $OAD = 180° - \beta$, lay off AD equal to the outer radius r_2, and join C with D, then the perpendicular erected at the middle M of CD will intersect AO at the center O of the arc AB of the vane. We see from the equality of the troughs OBC and OAD, whose sides are respectively equal, that the angle $OBC = OAD = 180° - \beta$, from which follows that the vane forms the angle β with the tangent to the wheel at B. Of course the centers of all the blades lie in a circle drawn through O concentric to C.

The experiments made by *Rittinger* on a fan whose vanes were curved back like AB, Fig. 58, so that the convex surface was in advance when in motion, gave in the most favorable case an efficiency of only 0.12. Moreover, these experiments showed that, owing to the small value of β (20°), the wheel had to receive a greater circumferential velocity than a wheel possessing vanes *curved inward* and ending radially like AB'', Fig. 60, in order to generate the same difference of pressure, a result which is explained by equation (11) of the preceding article.

According to *Rittinger's* experiments, therefore, it seems most advantageous to provide the fan with inwardly curved vanes which cut the outer circumference in the direction of the radius. Theoretically the number of vanes in a fan should be quite large in order that the motion of the air in the fan-passages may be as regular as possible; but the frictional resistance and the load on the shaft increase with the number of vanes. The number

is therefore seldom larger than 9 and is usually between 5 and 8. *Rittinger* gives *Dollfus's* rule that the distance between the adjacent vanes, measured at the outer circumference, may be equal to 0.21 m. [8.27 ins.]; under this supposition the number of vanes becomes

$$z=\frac{2\pi}{0.21}r_2=30r_2 \quad [z=0.76r_2].$$

It was mentioned above that, in order to utilize as much as possible of the absolute velocity of efflux

$$v_2=\sqrt{c_2^2+u_2^2-2c_2u_2\cos\beta},$$

the fan should have a *diffuser*. In suction-fans a diffuser is simply a continuation of the casing, which has the same width as the fan at the circumference of the wheel, but is sometimes larger at a distance from the latter. By placing guide-blades

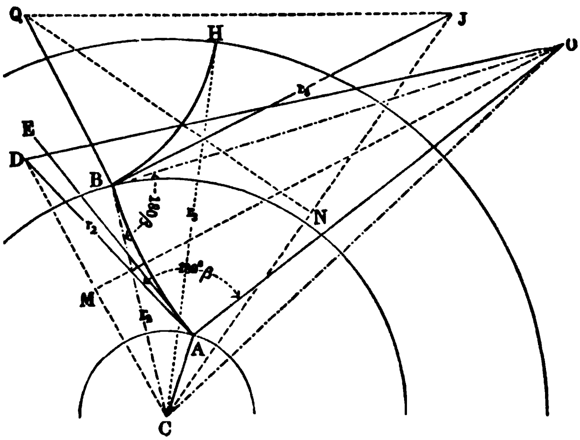

FIG. 61.

between the walls of the outlet, the action of the diffuser is improved and the guides must be placed so as to make with the outer circumference of the wheel the angle ϕ at which the air issues; in suction-fans these guides meet the outer circumference of the diffuser in a radial direction. If we also curve these guides to circular arcs, their center can be found as indicated above for vanes. Thus if BJ, Fig. 61, is the direction of

the air issuing from the fan, we lay off $BJ=CH=r_3$ equal to the outer radius of the outlet, draw CJ, and erect at its middle point N a normal which intersects at Q the perpendicular to BJ at B; the point Q will be the center of the diffuser-blade BH.

As regards the action of the diffuser we have (Fig. 58)

$$r_3l_3w=r_2l_2c_2\sin\beta=r_2l_2v_2\sin\phi,$$

where w is the velocity of the air which leaves the outer circumference at H radially, and l_3 is the width of the outlet at this point; the formula shows that w, and therefore the loss of head $\frac{w^2}{2g}$, becomes smaller the greater r_3 and l_3.

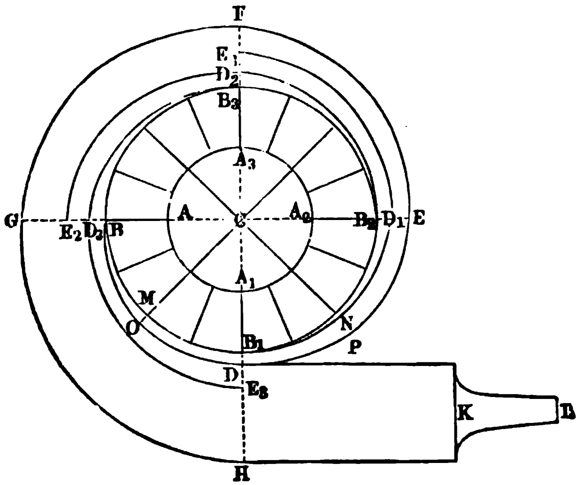

FIG. 62.

To design the outlet of fan-blowers, we first determine from the velocity w with which the air is to move in the pipe HK, Fig. 62, the cross-section $F=\frac{Q}{w}$ for this pipe. If we make this cross-section rectangular at DH with a width l_3 of the wheel, the height at this place will be

$$DH=a=\frac{Q}{l_3w}.$$

If the casing at point B is as close as possible to the wheel, we must, as in centrifugal pumps, give a spiral or involute

profile *BDEFGH* to the outlet in order that the air may have its direction gradually changed to that of the air-pipe *HK*. For a better guiding of the air within the outlet we may employ a few corresponding guides, as $B_1D_1E_1$, $B_2D_2E_2$, etc.

Example.—A fan is to deliver 1 cu. m. [35.3 cu. ft.] of air at a pressure of 0.2 m. [7.87 ins.] of water; what dimensions and what number of revolutions must be given to it?

If we suppose the air to approach with a velocity $v_0=10$ m. [32.8 ft.], and suppose a suction-opening in each side of the casing, and consequently that each side receives $\frac{1}{2}Q=0.5$ cu. m. [17.65 cu. ft.] per second, we obtain for the radius of the suction orifice

$$r_0=\sqrt{\frac{\frac{1}{2}Q}{\pi v_0}}=\sqrt{\frac{0.5}{31.4}}=0.125 \text{ m. } [4.92 \text{ ins.}].$$

If we assume the inner radius r_1 of the wheel equal to $1.2r_0$, we have

$$r_1=1.2\times0.125=0.15 \text{ m. } [5.9 \text{ ins.}].$$

Provided the vanes end radially at the outer circumference of the fan, we now have for the velocity of the outer periphery the value given by equation (8^a),

$$u_2=\sqrt{g\varepsilon(h+z)+\frac{w^2}{2}}.$$

To determine the values z and w in this expression, let us assume that the fan has an efficiency of 30 per cent.; we can then place

$$\eta=\frac{h}{h+z+\frac{w^2}{2g\varepsilon}}=0.30,$$

from which comes

$$g\varepsilon(h+z)+\frac{w^2}{2}=\frac{g\varepsilon h}{0.30}.$$

From this we obtain

$$u_2=\sqrt{\frac{g\varepsilon h}{0.30}}=\sqrt{\frac{9.81\times800\times0.2}{0.3}}=72.3 \text{ m. } [237. \text{ ft.}].$$

Now if we assume the outer radius of the wheel to be $r_2 = 0.5$ m. [19.69 ins.], the number of R.P.M. is given by

$$n = \frac{60 \times u_2}{2\pi r_2} = \frac{60 \times 72.3}{3.14} = 1380.$$

The half-width l_1 of the wheel at the inner circumference, when the entrance velocity v_1 is assumed equal to v_0, becomes $l_1 = 0.35 r_1 = 0.053$ m. [2.09 ins.]. The angle α made by the first element of the vane with the inner circumference of the wheel is determined by

$$\tan \alpha = \frac{v_1}{u_1} = \frac{10}{\frac{15}{50} \times 72.3} = \frac{10}{21.7} = 0.461;$$

hence $\alpha = 24° \ 45'$. The power needed to drive the wheel can be assumed equal to

$$L = \frac{Q\gamma_0 h}{\eta} = \frac{1000 \times 0.2}{0.3} = 667 \text{ m.-kg.} = 8.9 \text{ H.P.}$$

APPENDIX.

American Blowing-engines. — The improvements in blowers in the past twenty years have been chiefly in valves and valve-mechanisms. The purpose of these improvements has been the reduction of the clearance spaces and the quicker and more positive opening or closing of the valve at the proper time. The first of these improvements to be mentioned is the Gridiron Valve manufactured by the Southwark Foundry and Machine Company. The valve is shown in Fig. 63, and possesses many excellent features. This valve gives a large port-opening as soon as it begins to open, and requires very little motion for the full port-opening. The inlet-valve is made somewhat larger than the outlet-valve, to provide for the greater volume of air before compression.

The method of operating the inlet-valve is by a cam, operated by the engine, and so does not differ from other slide-valves. The operation of the outlet-valve is, however, automatic, and depends upon the air-pressure in the blowing-cylinder.

The outlet-valve is attached to the piston of the small auxiliary cylinder shown in figure. When the pressure in the blowing-cylinder equals that in the receiver, air admitted from the blowing-cylinder to the auxiliary cylinder causes the valve to open. At the end of the stroke of the blowing-piston a small pilot-valve, operated by the main valve-gear, admits air from the receiver to the other end of the auxiliary cylinder, thus closing the main valve. By closing the valve before the blowing-piston has begun its return stroke no air flows back from the receiver into the cylinder. The quantity of air used to operate the valve is very small, and is far less than would be lost by using valves operated by the air-pressure directly.

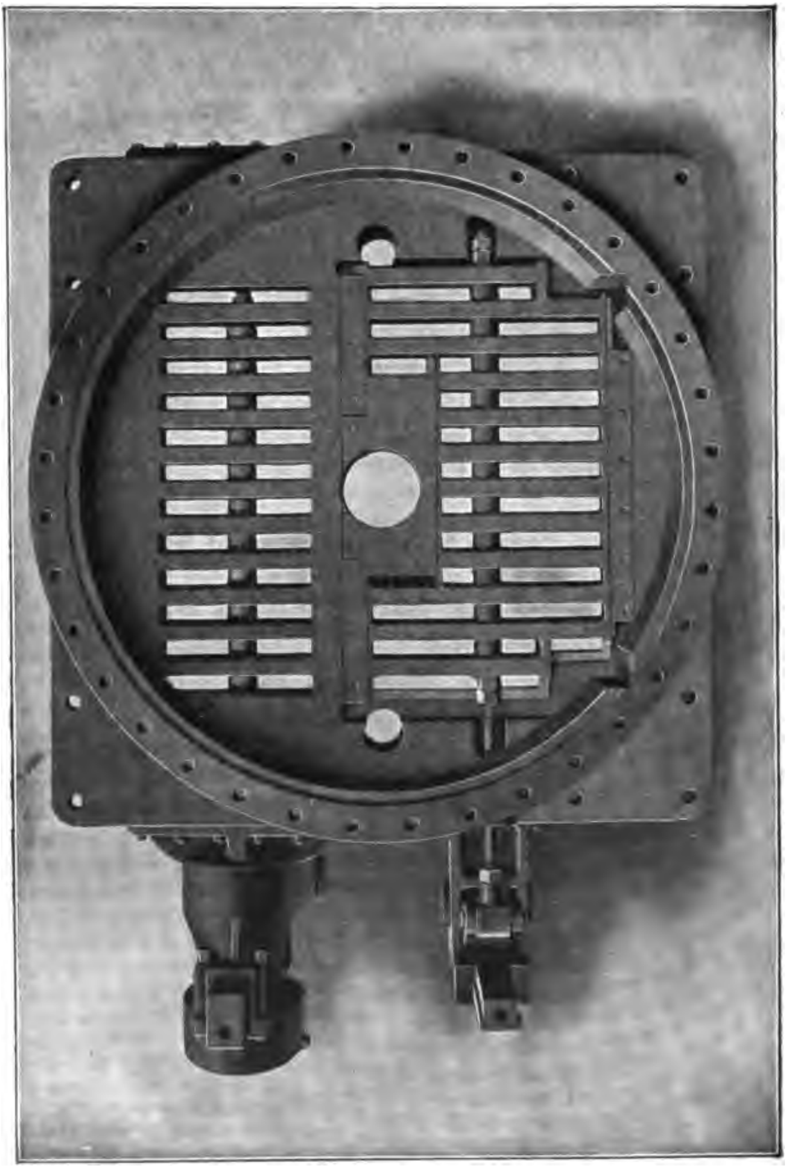

Fig. 63.

A very ingenious modification of the gridiron valve just described is a fan-shaped gridiron valve made by the same company for use on their horizontal blowing-engines. The method of operating these valves is identically the same as for the rectangular valves. Their feature is the large port-opening very well distributed over the head of the cylinder. Both types of these gridiron valves are designed to lift slightly from their seats when in motion, bearing on their back surfaces until the movement is completed, when they are again forced to their seats by the air-pressure. This reduces the wear on the valves and seats to a minimum, which is especially important in this case of the fan-shaped valves, where the relative velocities of the rubbing surfaces are not the same at all points and so tend to make the wear unequal. The fan-shaped valves are shown in Fig. 64, which shows the inlet valves only, the outlet valves being covered.

By the use of these valves and their operating mechanisms it is possible to run blowing-engines at such a speed as to get excellent results in the way of steam economy from the driving engine. Fig. 65 shows a set of indicator cards taken from a 44″ and 84″×66″ stroke horizontal-cross-compound engine driving two 84″ × 66″ stroke blowers equipped with these valves and valve-gear. It is possible to run these blowers at 90 revolutions per minute and still get very good results.

A very novel type of engine, but one that has proved highly satisfactory as a blowing-engine, is the disconnected compound shown in Fig. 66 and built by the Southwark Foundry and Machine Company. These engines are built either horizontal or vertical, and the only connection between the high-pressure and low-pressure engines is the steam-pipe or receiver.

There is a governor to control the admission of steam to the high-pressure engine only, as it has been found that the low-pressure engine will run at precisely the same speed as the high-pressure. Either the high-pressure or the low-pressure engine can be run separately, the only change required being the opening or closing of a few valves, thus giving a blowing plant of the utmost flexibility, and one that affords every facility for making repairs without shutting down the entire plant.

Blast-furnace Gas-engines.—Among the latest developments in blowing-engines is the application of gas-engines as a driving power for the blower. Gas-engines can be constructed to give a

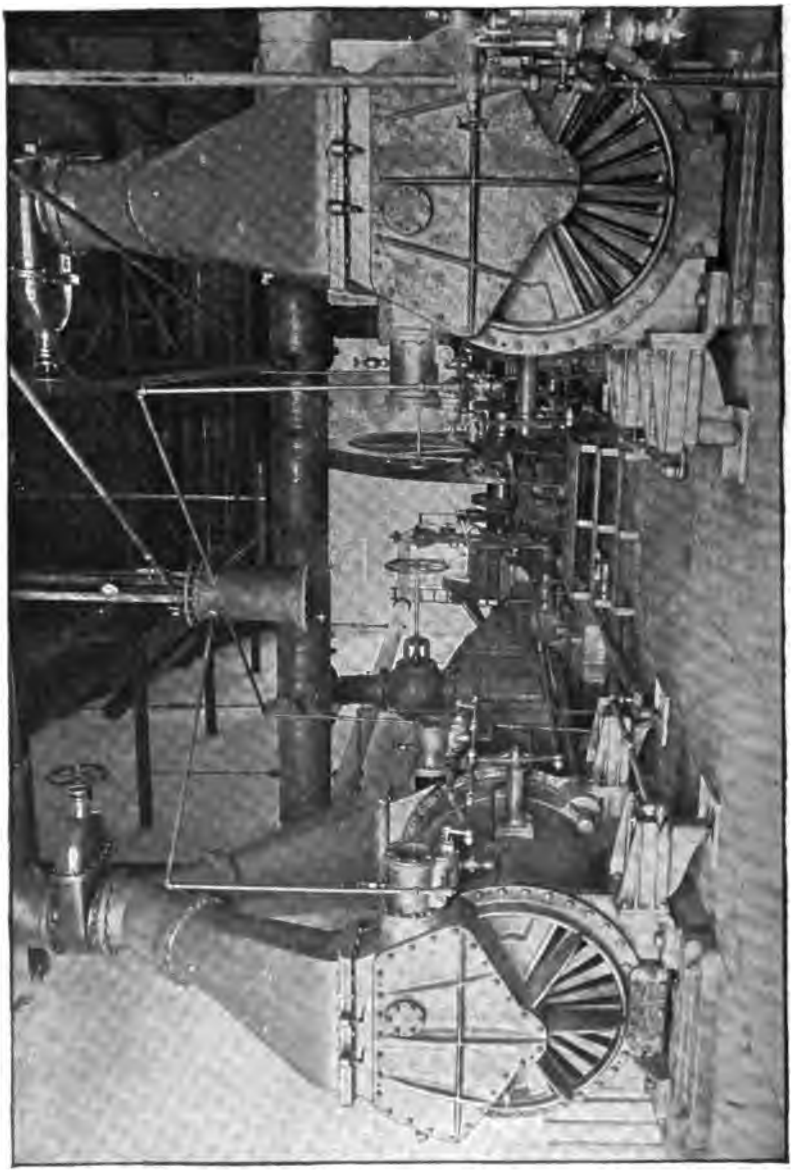

Fig. 64.

far greater thermal efficiency than steam-engines, and, as it has recently been found possible to use blast-furnace gas as a fuel gas in engines, it is probable that engines of this type will be very generally used. Fig. 67 shows a blowing set built by the Southwark Company, in which the blowers are driven by blast-furnace gas-engines. It is one of sixteen built for the same company.

It is estimated that by the use of these gas-engines the blast-furnace will not only furnish power for the blowers, but that

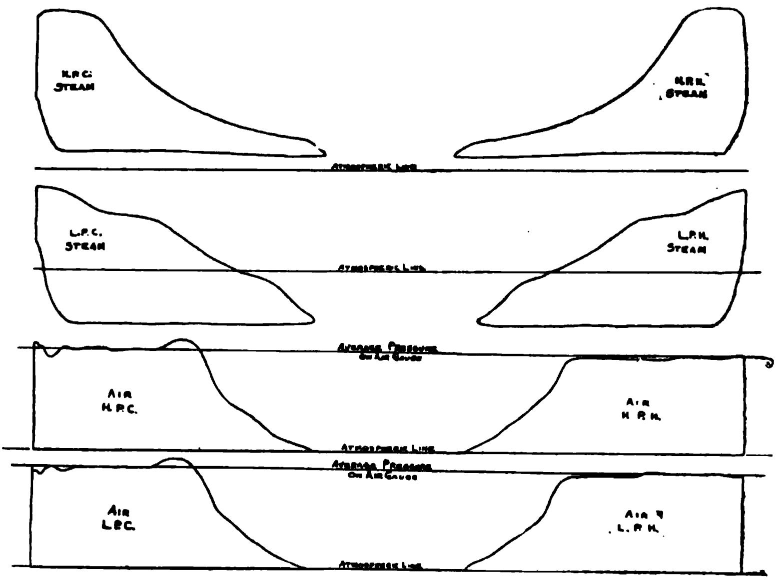

FIG. 65.

considerable power will be available for other uses. This means a great saving over the method of using steam-engines and boilers, as by that method the waste heat from the furnace did not furnish steam enough for the blowing-engines.

In Fig. 68 is shown a very commonly used type of blowing-engine as built by the Southwark Company, while Fig. 69 shows a duplex blowing-engine of the same type as built by the Allis-Chalmers Company. These engines are very frequently used because they combine the advantages of small height and small floor-space. The principal objections to them are the very long, heavy cross-head, the double connecting-rods and the inconvenience

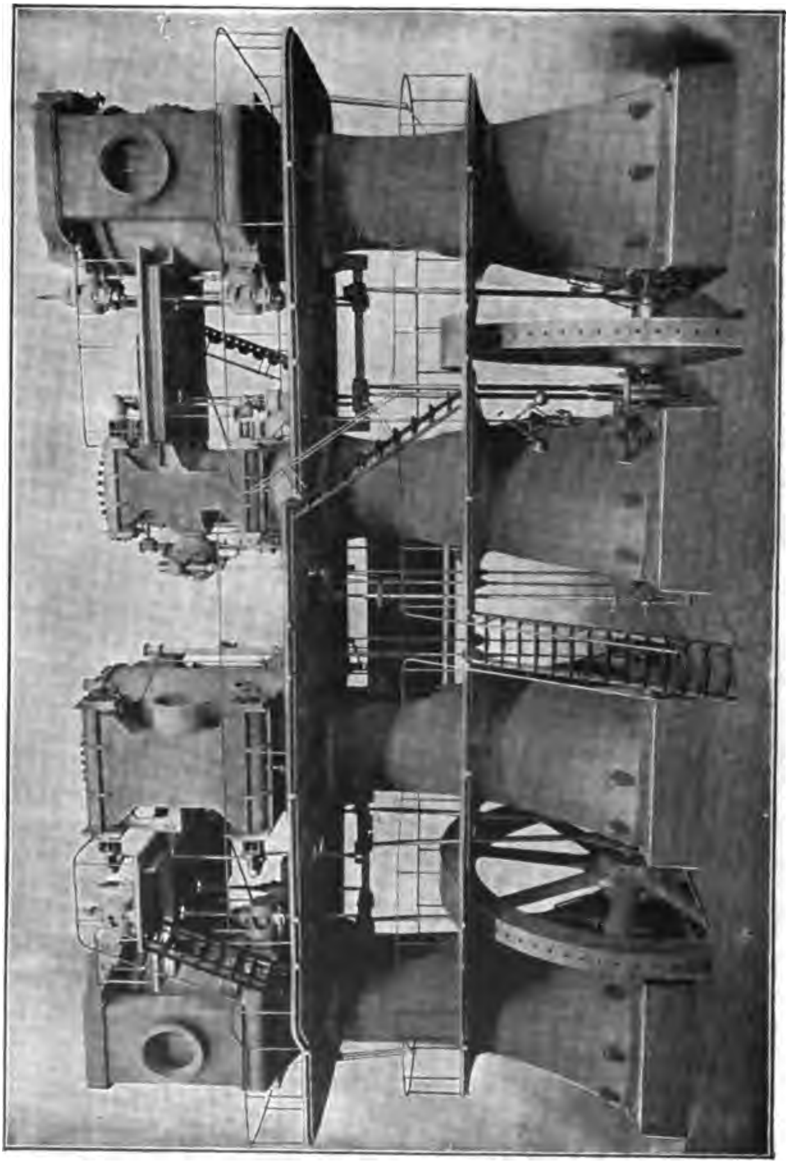

Fig. 66.

FIG. 67.

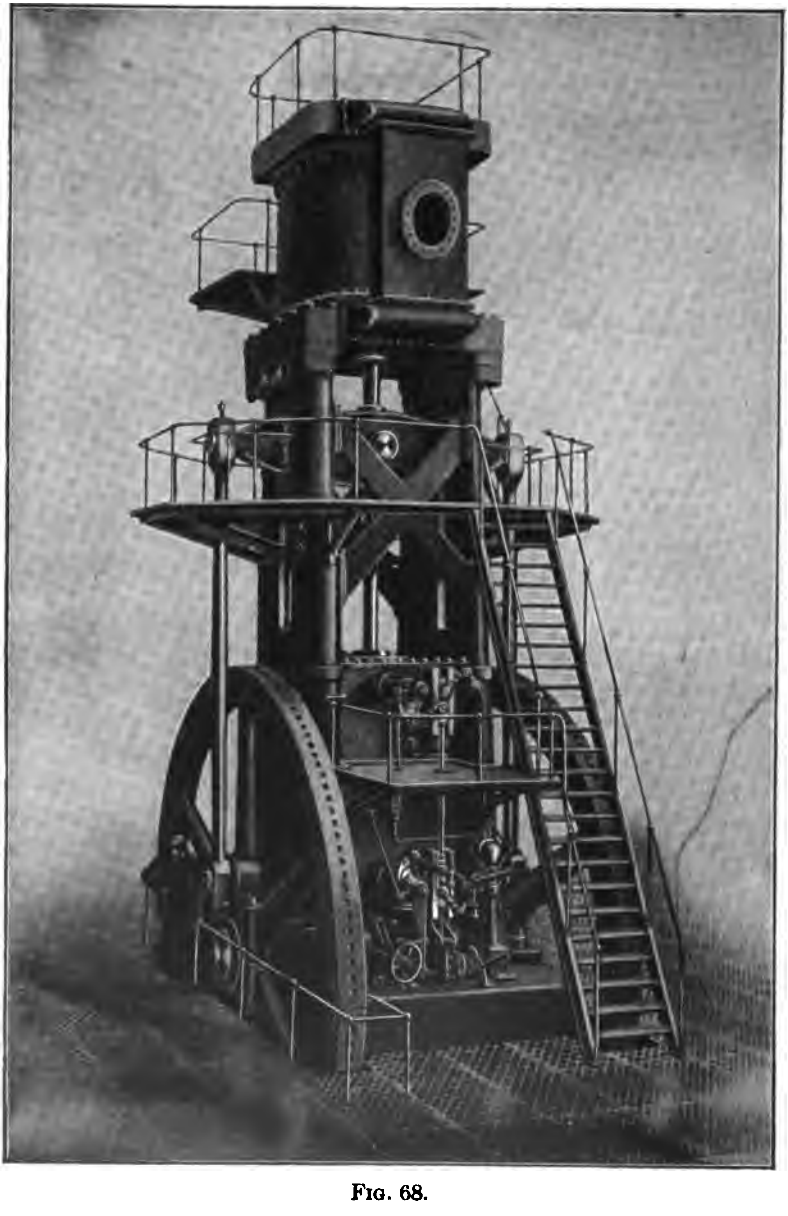

Fig. 68.

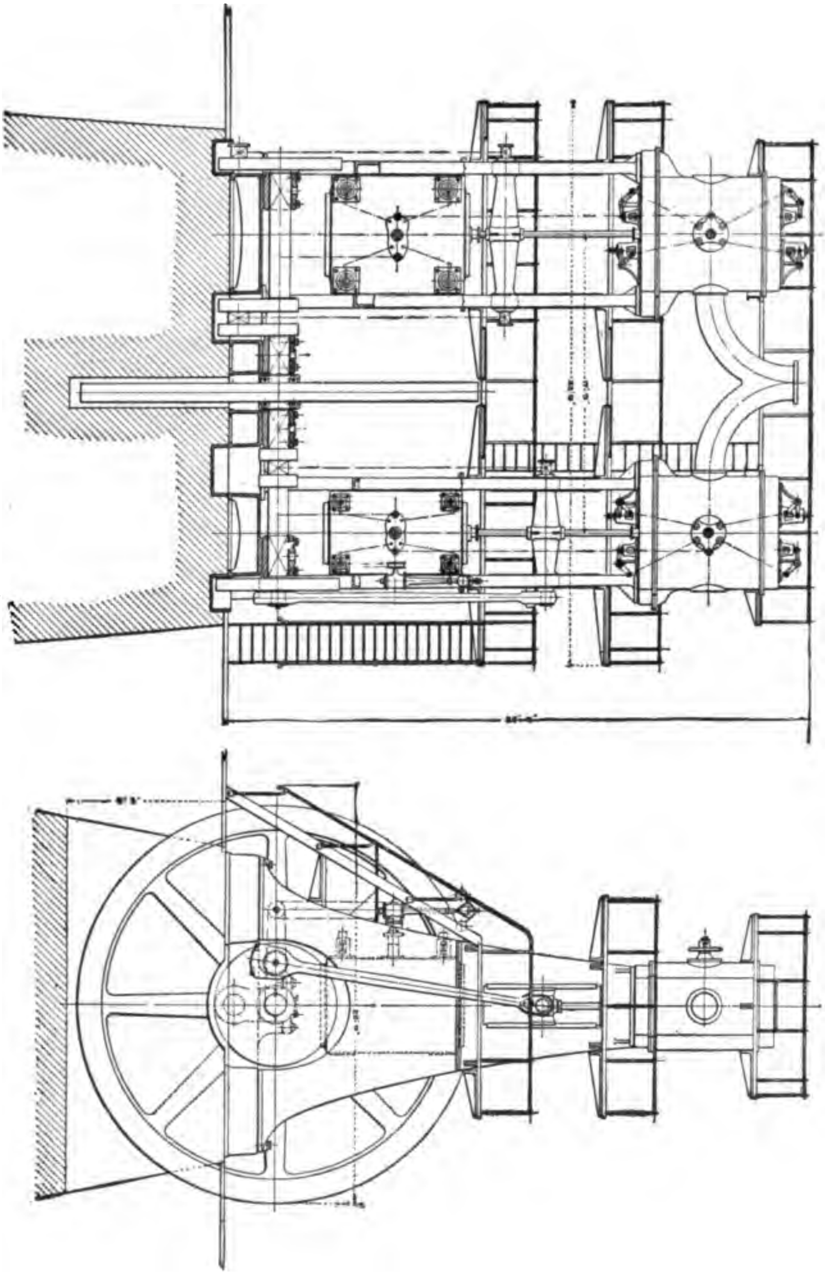

FIG. 69.

experienced in making repairs to the steam-cylinders which are located between the housings.

Compressors.—The demand for better and lighter air-compressors has led to great advances in the past twenty years in this particular line of air-machines. The greatest improvements have necessarily been in the valves and in the methods of operating them, so that higher piston-speeds and a greater number of revolutions could be used.

Fig. 70 shows the cylinder of an air-compressor fitted with the Riedler air-valves as built by the Allis-Chalmers Company. These valves are operated by the cams shown at 14 and 15, and the amount of opening can consequently be made as great as desired. On account of the valves being controlled by the engine mechanism, the compressor can be run much faster than if it were fitted with clack-valves. The form of the valves makes possible a very small clearance. The stems of these valves are attached to plungers working in dash-pots, which effectually check any vibrating tendency of the valve. The friction of the air in the ports is very small, on account of the large lift that is possible with these valves.

The names of the parts shown in Fig. 70 are as follows:

13. Cam-plate.
14 and 15. Cams.
16. Cam-plate lever.
17 and 18. Cam-rollers.
19. Slide.
20, 21, 22, and 23. Valve-rods.
24 and 25. Valve-forks.
26, 27, 28, and 29. Valve spindle-levers.
30, 31, 32, and 33. Valve brackets.
34 and 35. Cylinder-drains.
36. Inlet to water-jackets.
37. Water-jacket piping.
38. Cam-plate connecting-rod.
39 and 40. Tie-rods.
41. Water-jacketed air-cylinder.
42 and 43. Water-jacketed cylinder-heads.
44. Inlet-valve bonnet.
45. Outlet-valve bonnet.

In Fig. 71 is shown a very compact form of air-compressor or blowing-engine which is built by the Allis-Chalmers Company. The peculiarity of this machine is the triangular link or bell-crank which connects the crank to the two short connecting-rods of the engine, by means of which mechanism it is possible to place the air and steam cylinders very close together and very close to the floor. The advantage this machine has over the type shown in

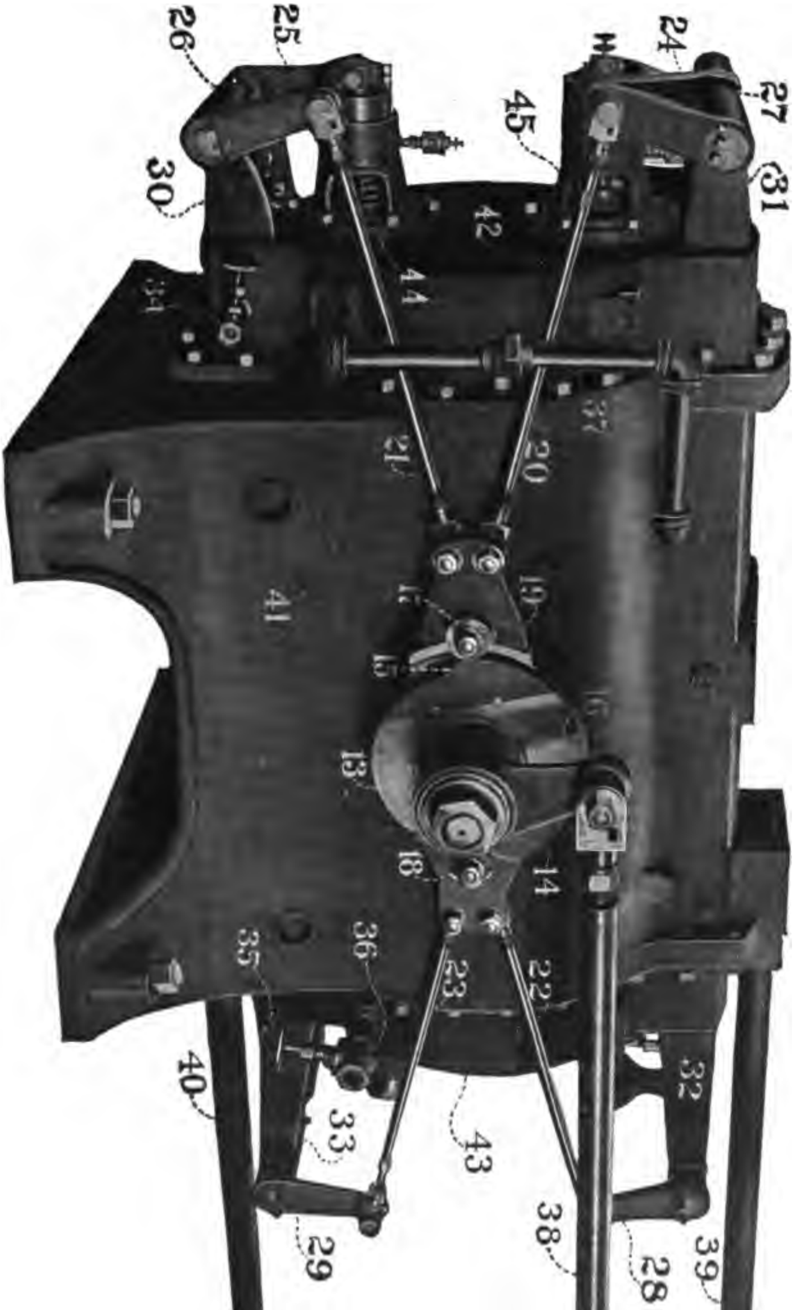
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Fig. 70.

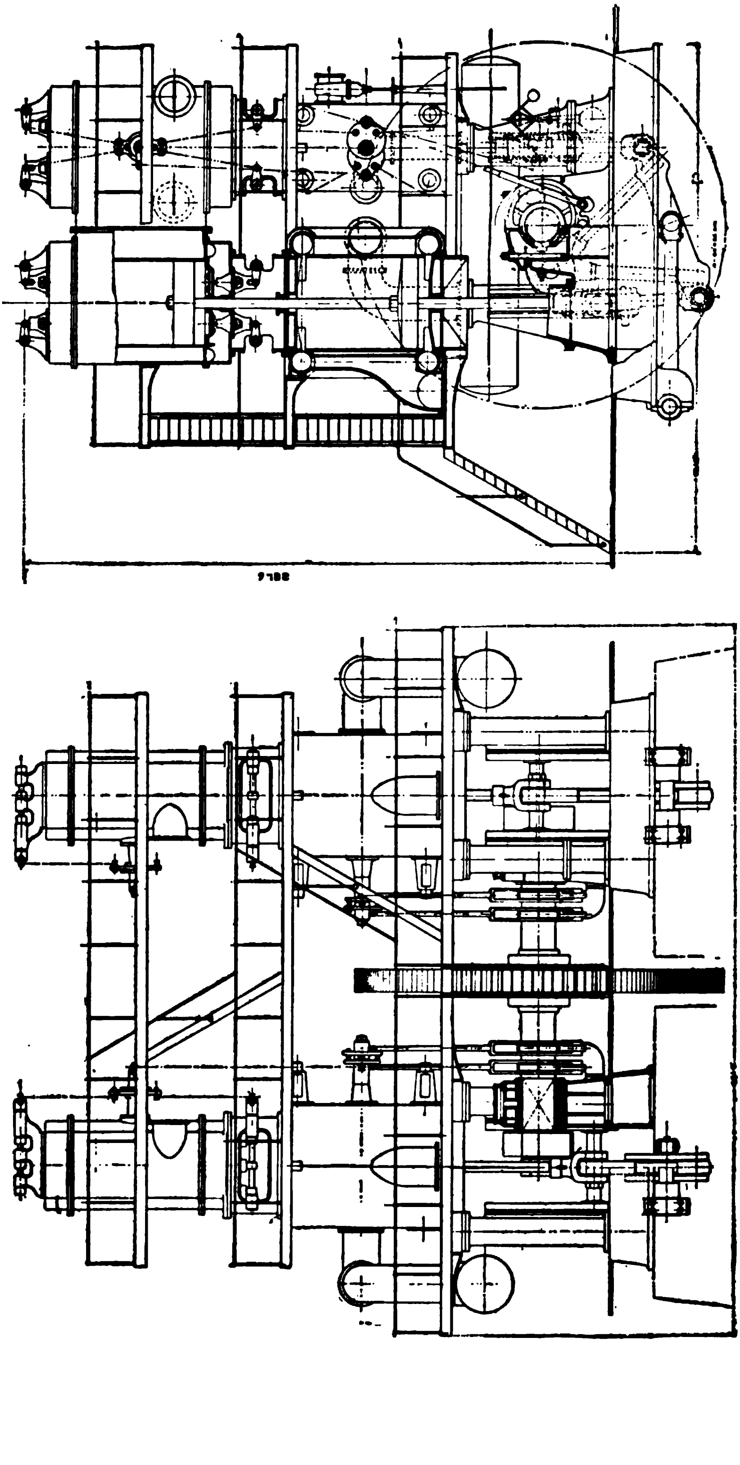


Fig. 71.

Fig. 68 and Fig. 69 is that it avoids the long cross-heads and the double connecting-rods.

In some instances it is necessary to use air-compressors driven by high-speed engines, so as to provide for easy transportation. Here the gas-engine becomes an important factor. By reason of the high speed and very large mean effective pressure possible, they can be quite small as compared with a steam-engine of the same power. Hence they are decidedly useful for furnishing power in places that are difficult of access or where the fuel must also be transported. As these engines do not work satisfactorily at speeds less than 200 revolutions per minute, they can only be used direct-connected on very high-speed compressors. Connecting them to the compressor by gearing reduces the efficiency and simplicity of the machine so much that it should be avoided whenever possible.

To meet the demand for a high-speed compressor, the Allis-Chalmers Company have developed what they call the Express air-compressor, capable of working efficiently at speeds as high as 200 to 300 revolutions per minute. The valves for these high-speed compressors are called Express valves, and one is shown in section in Fig. 72. This is a delivery-valve and opens inwardly by the pressure of the air in the cylinder. A lip on the outside of the valve furnishes a cushion for the valve on opening. The spring buffer on the piston closes it at the end of the stroke, and as this is done while the piston is coming to rest, the motion is slow.

The names of the parts shown are as follows:

A. Delivery-valve.
B. Outer half of valve-guide.
C. Inner half of valve-guide.
D. Spring rings.
E. Valve-seat.
F. Valve-seat spring.
G. Valve-cover.
H. Cylinder-head.
I. Air-passages.
J. Cylinder.
K. Water-jacket.
L. Piston.
M. Buffer-spring.
N. Buffer-stud.
O. Buffer-stud nut.
P. Buffer-plate.
Q. Lock-nut.

Among other valves for air-compressor service, the Ingersoll-Sergeant piston inlet-valve deserves mention. This valve is unique in American air-compressor practice. The essential

features and general construction are shown in Fig. 73. The piston inlet is a tube of ample cross-section, screwed into the hollow air-piston and travelling in a stuffing-box in the rear cylinder-head. The piston is a hollow casting furnished with the usual

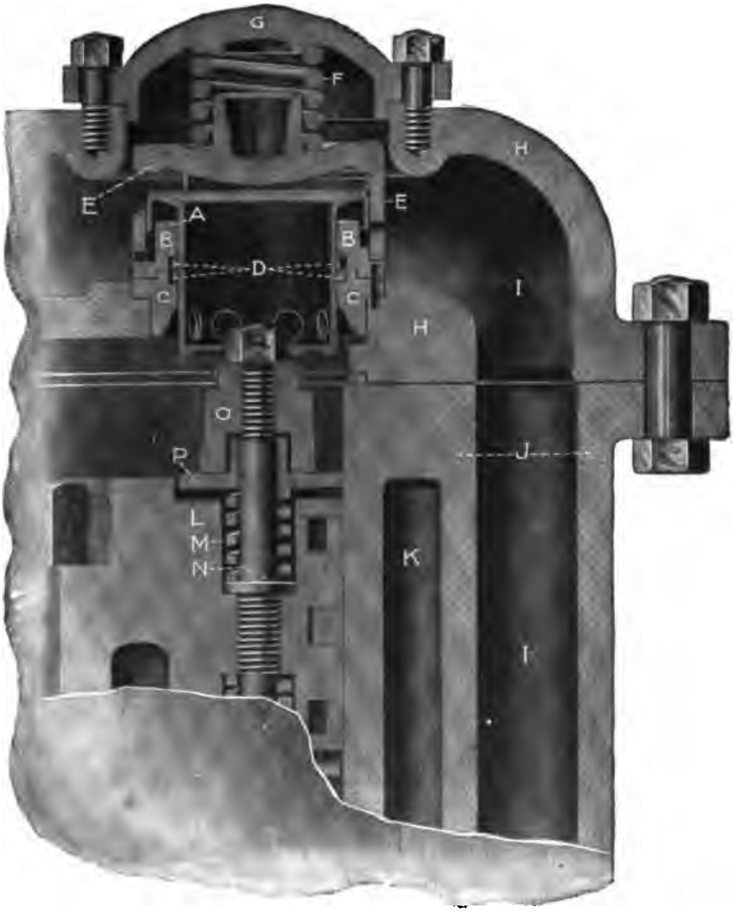


FIG. 72.

segmental rings, and having on each face an annular opening which is the air intake to the cylinder. One or both edges of this opening are finished for valve-seats, according as the valve is single- or double-ported. The valve *G* is forged from a solid billet of open-hearth steel, turned to accurate surfaces on sliding contacts and face. Hardened steel pins, inserted beneath the

piston-rings, enter the guide-slots in the valve and hold it in position. There are no springs used in this valve. The operation of the valve is as follows: Assume the piston at the extreme right end of the cylinder, just starting to the left. The left-hand valve will be open and the right-hand valve closed. As the piston moves, the inertia of the movable valves causes the left-hand valve to close instantly and the right-hand valve to open. At the same time the pressure falls in the right-hand end of the cylin-

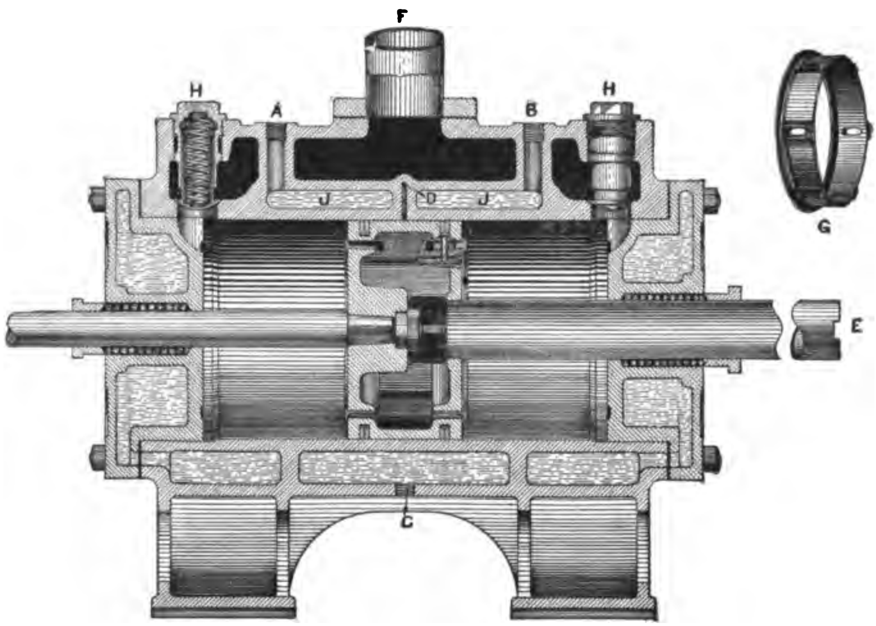


Fig. 73.

der and air rushes in through the piston inlet-tube, the hollow piston and open valve into the cylinder. The air in front of the piston being compressed, holds the leading valve firmly to its seat and the air is forced out through the discharge-valves. At the left-hand end of the stroke the valves are operated in the opposite way, the inertia of the right-hand valve closes it instantly, and the inertia of the left-hand valve, combined with the reduced pressure in that end of the cylinder, causes the left-hand valve to open, allowing air to enter through the inlet-tube, piston and valve. It is obvious that, since the valve cannot travel faster

than the piston, there is no tendency to close the intake-valve until the end of the stroke.

The features that are of special interest in this valve are the absence of any operating mechanism requiring additional power; the use of the inertia of the valve as its operative factor instead of a retarding influence; the large, unobstructed intake passage; the absence of springs, with their liability to breakage; the small clearance; the complete cylinder and head jacketing permitted through the absence of valves in the heads, making jacket-cooling

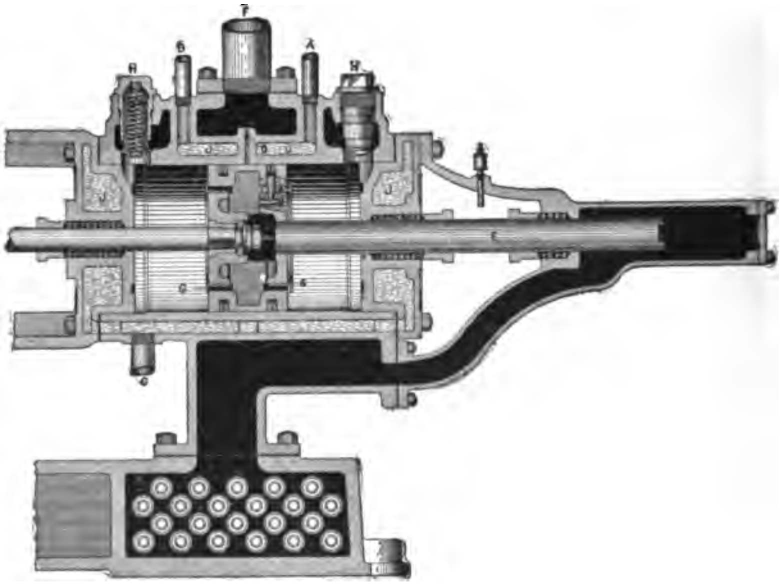


FIG. 74.

available where most effective; the high efficiency secured by large intake passage, ample port area, and small clearance; the function of tail-rod or back piston-guide served by the piston inlet-tube, reducing friction and wear between cylinder and piston and the readiness with which cool and clean intake-air may be secured by leading the piston inlet-tube into a conduit communicating with a suitable source of supply. The method of application of the piston inlet-valve to the high-pressure cylinder of compound or two-stage machines is shown in Fig. 74, which represents the high-pressure cylinder of a two-stage compressor carrying an intercooler in the sub-base. Where a separate inter-

cooler is used, a pipe from the intercooler is attached to the right-hand end of a bracket similar to that shown.

The Ingersoll-Sergeant poppet discharge-valves are shown in section and in position in Fig. 73, and Fig. 75 shows the separate details. The valve-cap is of gun-metal, and is screwed into the walls of the air discharge-chamber. It is accurately bored to serve as a guide for the valve. The spiral spring is wholly enclosed by valve and cap. The valve proper is of high-carbon steel, toughened in oil, annealed and ground to size. It will be noted that the valve is very light, every particle of superfluous metal being machined away to make as small as possible the retarding action of the inertia of the moving parts. Under ordinary circumstances the valve closes directly on a ground valve-seat in the cylinder-wall, but in special cases a separate valve-seat of

FIG. 75.

suitable material is furnished, fitted to a taper reamed hole in the cylinder-wall. This valve is very light, has wide bearing surfaces in its guides, preventing any tendency to spring or bind, has ample port area with small lift, and is very easily accessible for inspection or cleaning. This type of valve is suitable for very high pressures, having been used on machines compressing to 1500 pounds per square inch.

The Ingersoll-Sergeant Company also make a positive air-thrown valve which is usually furnished on Corliss compressors of large capacity where the conditions of volume, pressure, and fluctuating load render the use of such valves advisable. Fig. 76 shows these valves in place and in section. Both inlet and discharge valves are of the poppet type with large sleeves serving as guides. Axial valve-stems carry pistons which operate in the external cylinders. These cylinders are piped so that air at a pressure equal to that in the compressing cylinder operates to

hold the valves closed, while air at a much reduced pressure from the small auxiliary receiver shown operates to open the valves.

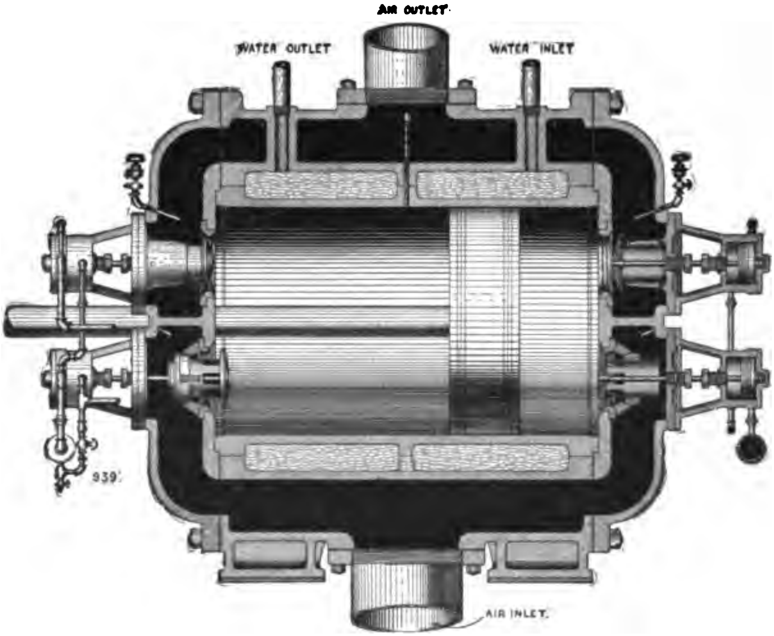


FIG. 76.

The pistons and cylinders further serve as a dash-pot, reducing the jar on seating of the valves. The admission and release of

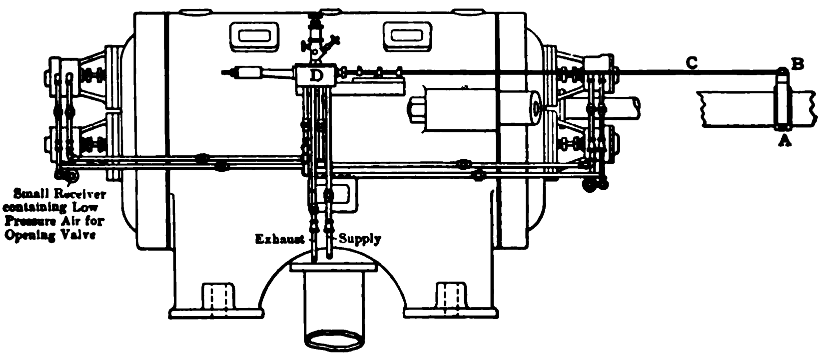


FIG. 77.

air furnished to the valve-cylinders is controlled by the small auxiliary slide-valve *D*, Fig. 77, mounted on the side of the cylinder.

This auxiliary valve is moved by a reducing motion from the main piston-rod. By means of a lost-motion device the valve is moved only at the end of the main piston-stroke and hence the amount of air required by the valve mechanism is very small. Assuming the piston at the right-hand end of the air-cylinder and just starting on the return stroke, the operation of the valve mechanism is as follows: Air-pressure equal to that in the compressing cylinder is acting on the right-hand discharge-valve piston, holding it tightly closed; an equivalent air-pressure holding the right-hand intake-valve closed has just been released by the auxiliary slide-valve; hence the suction has a tendency to open this valve, but its weight and friction must be overcome when the valve will open and remain open until the piston has completed its stroke. At the same time working pressure is made to close the intake-valve in the left-hand end of the cylinder and the pressure holding the discharge-valve closed is relieved, which allows the discharge to open just as soon as the pressure in the cylinder equals the receiver pressure. On the return stroke this action is exactly reversed. The results secured from the use of these valves are best shown by the accompanying indicator card, Fig. 78.

It is noticeable that the intake line coincides very well with the atmospheric line, showing ample port area and prompt opening of the valve; that the compression line starts on the atmospheric line, and that the discharge line is straight, being free from the irregularities due to chattering of the valves and choking of the ports. The principal advantages of this valve are the mechanical control secured at small expense of power; the rapid and complete opening and closing of the valves at the proper time; small clearance space and the simplicity of the mechanism. This valve can be used for high pressures and can be operated at any speed within the limits of the Corliss engine.

Regulating Devices.—When part of the load of an air-compressor is removed, or the demand for air is diminished, it is evident that the air-pressure will increase and blow off at the receiver safety-valve. This escaping air represents so much lost power, and some means should be provided to make the volume of air compressed proportional to the load on the machine. The Ingersoll-Sergeant unloader accomplishes this result, and is shown in plan and elevation by Fig. 79.

In operation under conditions of normal pressure, air under

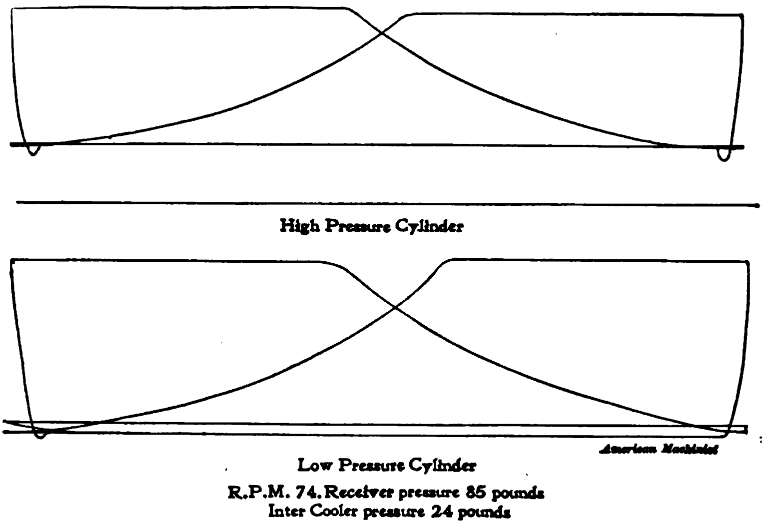

High Pressure Cylinder
American Machinist
Low Pressure Cylinder
R.P.M. 74. Receiver pressure 85 pounds
Inter Cooler pressure 24 pounds

FIG. 78.

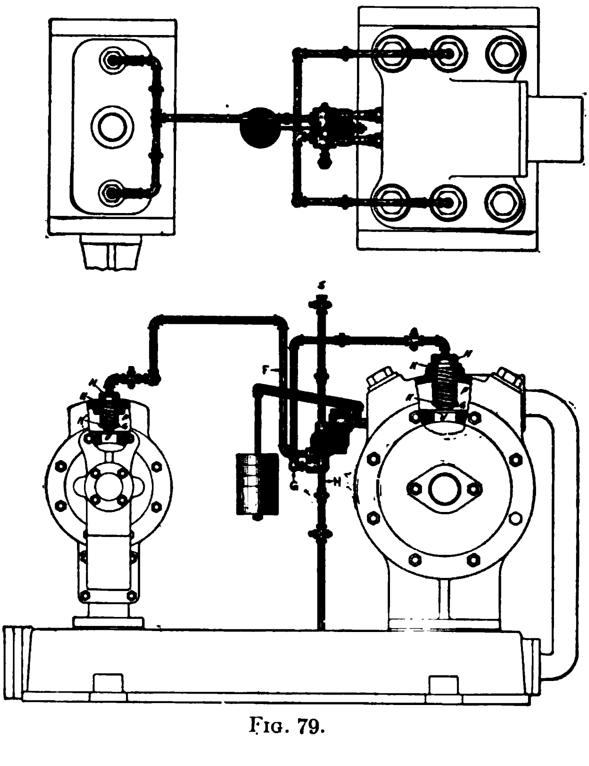


FIG. 79.

pressure is admitted through the pipes shown to the rear of one or more of the poppet discharge-valves on both ends of the cylinder, which are then held to their seats by the springs. When normal pressure is exceeded, receiver pressure acting on a small piston lifts the lever and weights and releases the air-pressure on the valves, which are then forced open by the pressure in the cylinder. Air under pressure is thus admitted to both ends of the cylinder and the piston travels under balanced pressure, the intake-valves being held closed by the pressure. When normal pressure is restored, the weights cause the lever and unloader piston to descend, admitting air under pressure to the discharge-valves, which resume their function and compression begins. This device may be adjusted to any pressure by changing the weights.

The illustration given shows the unloader attached to a compound machine, unloading both cylinders. It is equally applicable to a simple or a duplex compressor. The device as described will control a belt- or power-driven compressor, but when applied to a steam-driven machine it is used in connection with a steam regulator by means of which, at the instant of unloading, the steam-supply is throttled and the speed of the compressor reduced. This combination makes the steam consumption of the compressor proportional to the load upon it. Used on a power-driven machine, the unloader makes the demand for power proportional to the demand for air.

Another device for the automatic governing of power-driven compressors is the Ingersoll-Sergeant choking controller. This is a device for making the volume of intake-air proportional to the demand for compressed air. It is applied to larger machines than the unloader just described, for the reason that the sudden and complete unloading of a large compressor is likely to cause destructive shocks. Fig. 80 shows the controller applied to a piston inlet-machine.

In this device a multi-ported piston-valve inserted in the air-intake conduit is opened or closed by air-pressure from the receiver acting against the weights on the lever. Pressure in excess of the normal lifts the lever, partially or wholly closing the intake conduit and causing the piston to travel in air that is at a very reduced pressure. The capacity of the compressor is reduced by this means to a varying extent, and the power required is reduced in proportion. To prevent shocks due to too rapid opera-

tion of the valve, an oil-damping device is provided which compels gradual movement. This device is applied only on the low-pressure intake of the compressor, as its effect is carried through to the high-pressure cylinder in the reduced output from the low-pressure cylinder. Adjusting-screws permit setting of the con-

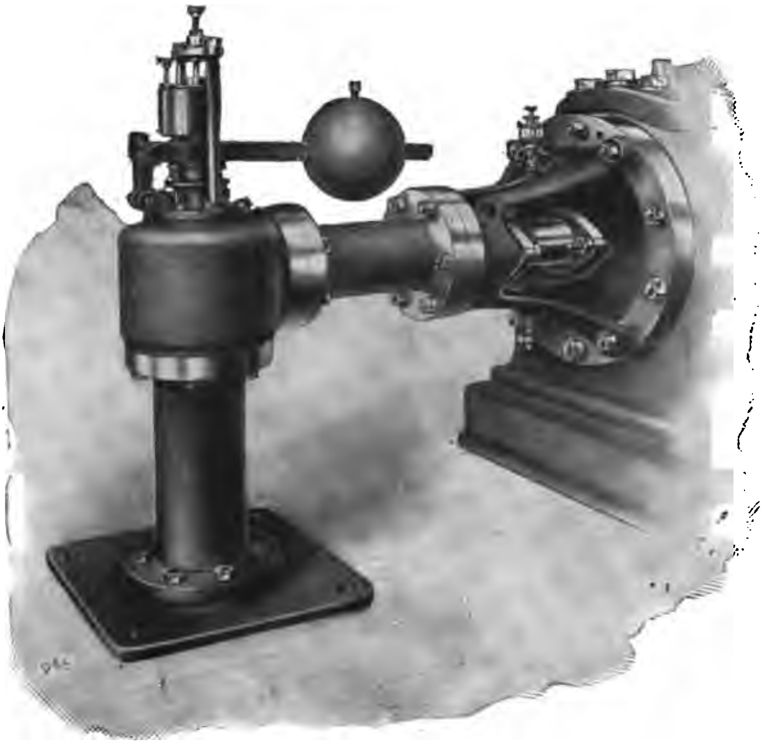


FIG. 80.

troller for any degree of throttling or unloading that may be necessary.

Fans.—The actual changes that have been made in fans in the past twenty years are very slight. There has been an increase in their size and in the velocity at which they have been driven, and by more careful attention to details the efficiency has been increased. The greatest change has been in the application of fans and in their almost universal substitution for all other kinds of blowers where air at a low pressure is required. So universal has the use of fans become that it is difficult to find any indus-

try in which they are not or could not be used with advantage. Probably no change is more noticeable than the completeness with which the fan-blower has supplanted the bellows for blacksmith's forges, and Fig. 81 is an illustration of a blacksmith's forge as made by the Buffalo Forge Company. The fan-blower shown on this forge has a capacity far greater than that of a large bellows, and can deliver air at a much higher pressure, thus materially

FIG. 81.

increasing the capacity of a hand-forge and doing heating more quickly and efficiently than is possible with a bellows.

Forced Draft.—Among the most important uses to which fans have been applied is that of furnishing forced draft or induced draft to boiler-furnaces. The reasons for using mechanical draft are different for the different cases, but it is usually done to increase the capacity of the boiler plant over its possible capacity using natural draft. Sometimes it is done to burn fuels that cannot be burned without a draft stronger than can be furnished by a chimney of reasonable height. The styles of fans used for this purpose are varied to suit the circumstances. Fig. 82 shows a type that is frequently used when the power for driving the blower is taken from a shaft. This blower is called by its builders, the Buffalo

Forge Company, a "B" volume blower. As these blowers rarely have fan-wheels exceeding three feet in diameter, they are run at a high number of revolutions, usually from 400 to 1000 revolutions per minute.

Steel-plate Fans.—One of the most popular types of fans for mechanical draft is illustrated by Fig. 83, which shows a steel-plate fan built by the B. F. Sturtevant Company. These fans can be varied to suit almost any case where a large volume of air at

FIG. 82.

a small pressure is required. The style of fan shown in Fig. 83 is built almost entirely of mild steel, the casing or housing being made of steel plate and the fan-wheel, shown in Fig. 84, being made of steel plate and angles, except the hub, which is of cast iron. These fan-wheels are frequently used in either wooden or brick housings, although the steel-plate housing combines lightness and strength much better than either of the others.

As the air-pressure required for forced draft rarely exceeds one and one-half inches of water, and should never exceed three or four inches, these fans are very suitable for this purpose. At these pressures the peripheral speed of the fan is comparatively low, and consequently they run easily and require very little atten-

tion, which makes the increased power of the boilers cost very little.

Having determined the peripheral speed required for a fan

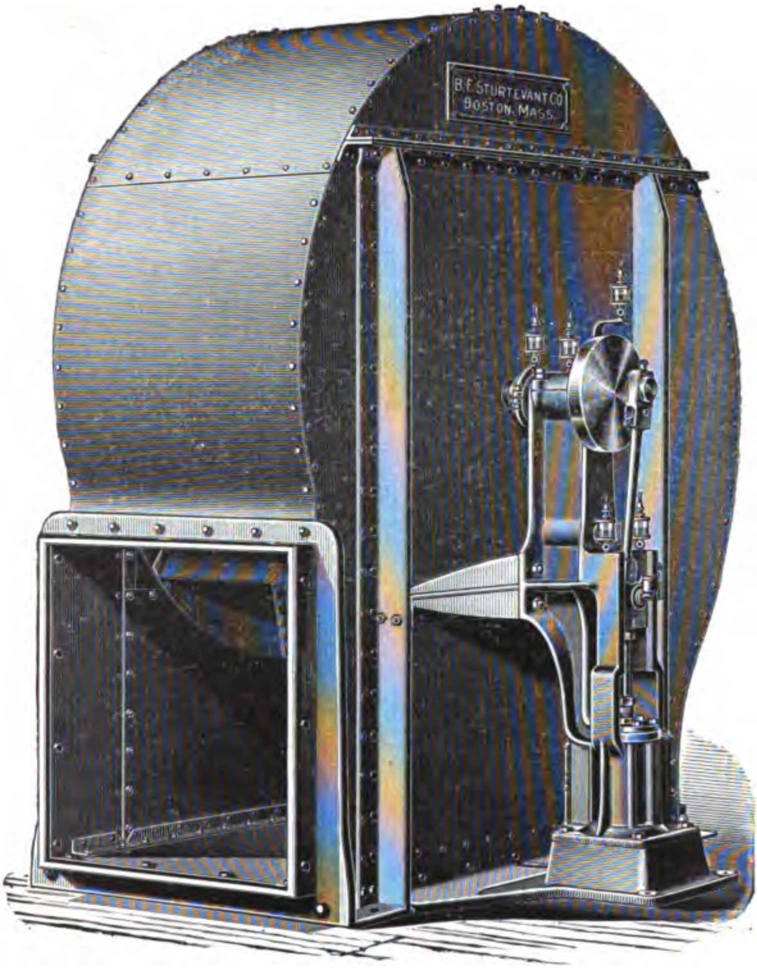


FIG. 83.

to give a certain air-pressure, the quantity of air delivered can be made as large as needed for any particular case, by making the fan wide enough, as the volume delivered at any given velocity depends directly on the width of the fan. Where great quantities of air are required, two fans of the same diameter are often installed

because one fan of sufficient width would necessitate an unduly heavy construction, and furthermore the duplex fans make a more flexible blowing-plant. When duplex fans are installed, they can be driven by two separate engines and a coupling provided so that either engine can drive both fans whenever it is necessary to make repairs on an engine. The fans themselves

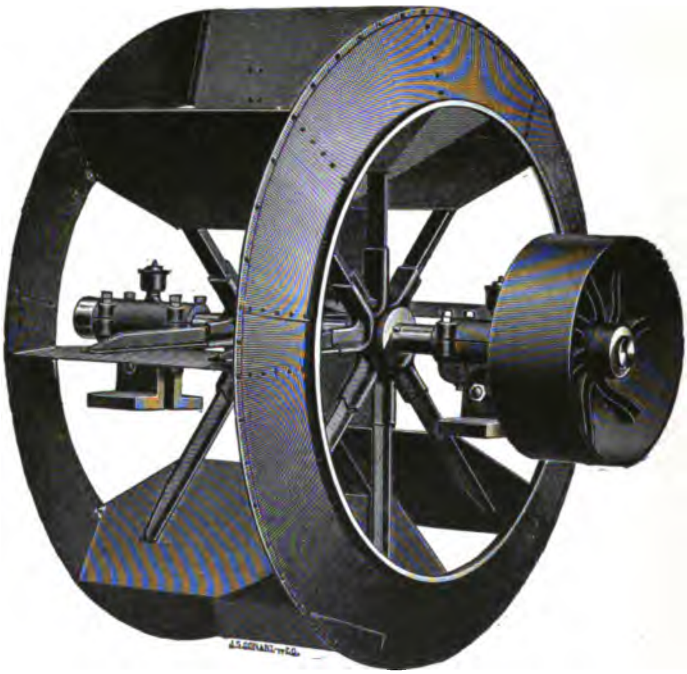


FIG. 84.

need repairs so seldom that another unit would never be installed on account of repairs to them.

Another use to which fans have been applied very extensively is for heating and ventilation. All large buildings and some small ones are now furnished with some mechanical system for thoroughly changing the air throughout the building, which is usually done by blowing in fresh air, either heated or cold as desired. The types of fan most used for this purpose are the steel-plate fan, Fig. 83, and the cone fan shown in Fig. 85, another of the B. F. Sturtevant Company's products.

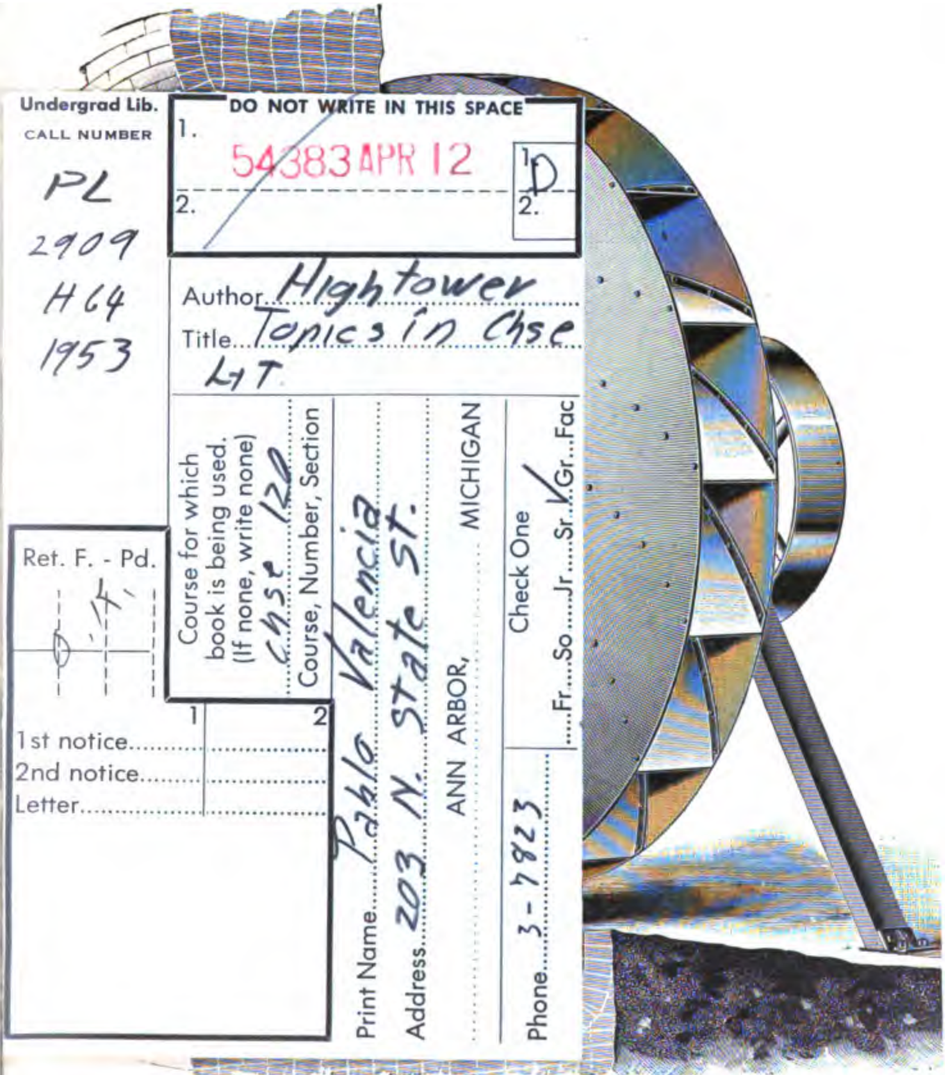


FIG. 85.

Disc Fans.—When the ventilation is to be secured by an exhauster and only a slight air resistance has to be overcome, the disc fan is usually the most desirable type to use. Fig. 86 shows a disc fan made by the American Blower Company, capable of exhausting from a compartment and also delivering the air against a pressure of from one to two ounces per

Fig. 86.

square inch. The object of using such large discs for holding the blades at the center is to prevent a current of air from flowing back through the fan when working against a pressure. Disc fans are somewhat more efficient than the other types for moving large volumes of air at very small pressures, and are excellent for such purposes as the removal of smoke, noxious fumes or gases, and in connection with drying apparatus.

Mine Ventilation.—When fans are to be used for ventilating purposes where it is necessary to maintain a comparatively large difference of pressure, as in the ventilation of mines, they are usu-

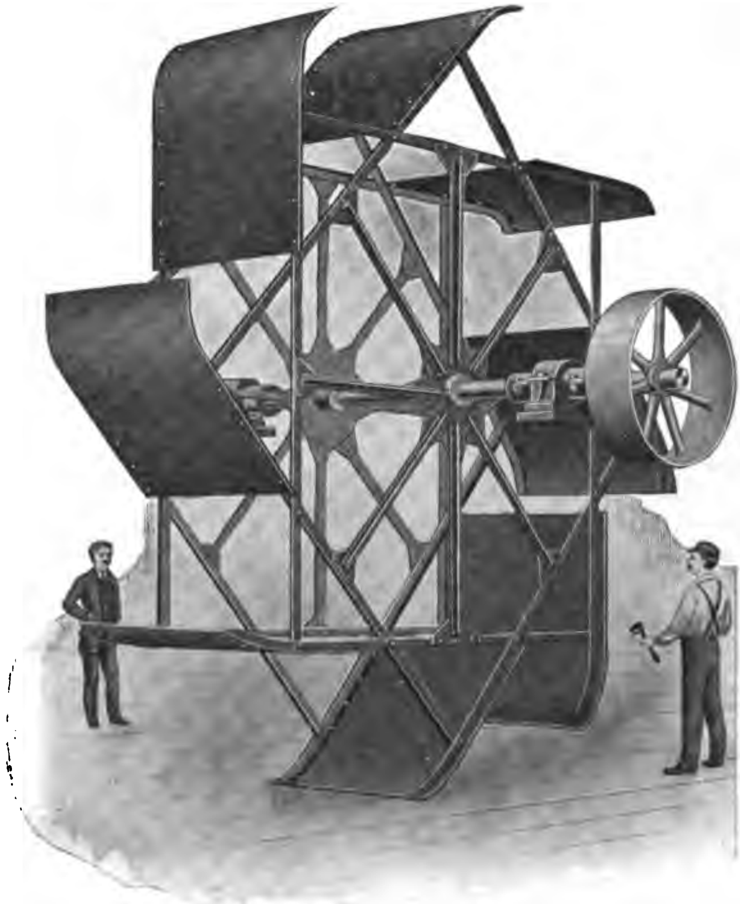


FIG. 87.

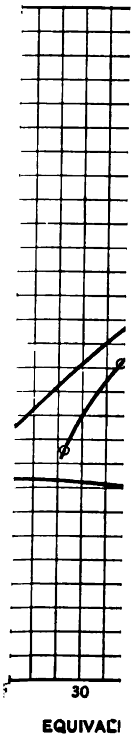
ally of the enclosed type having peripheral discharge with a diffuser. As the power needed for driving large fans for mine ventilation is large, it is customary to use the most efficient fan-wheels which can be obtained. Fig. 87 shows a Guibal type fan-wheel constructed by the American Blower Company and used for mine ventilation. The Guibal fan has been found to be the most efficient fan for

the ventilation of mines and tunnels. To obtain the greatest efficiency, however, care must be taken to have the shape of the casing a suitable spiral, so that the fan will discharge constantly during its revolution.

In Plate I the results of a series of tests are given which were made on a fan of this type but having the blades curved in the opposite direction to those shown in Fig. 87. These tests were made at a mine belonging to the Mineral Railroad and Mining Company at Shamokin, Pa. A fan which was identically the same as this one, with the exception that the blades were curved away from the direction of rotation, as shown in Fig. 87, was also tested and gave twenty-five per cent less discharge under the same conditions and at the same speeds.

When the quantity of air which a fan is to discharge varies, it is advisable to have a shutter in the discharge-pipe to regulate the size of the discharge-opening to correspond to the quantity of air required. In the case of certain fans which were not provided with shutters it was found that currents of air entered the fan through the discharge-opening when they were run at reduced speed. This is prevented by a shutter, which consequently increases the efficiency of the fan at these low speeds.

For a detailed description of various fans used in mine ventilation see an article by Mr. R. V. Norris on Centrifugal Ventilators in the Transactions of the American Institute of Mining Engineers for 1891.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EQUIVACI

Many of the results that have been obtained by experimenters from tests of centrifugal fans are of little or no value, because of inaccuracies in the method of taking observations to determine the quantity of air delivered. In 1884–5 the Prussian Mining Commission performed a series of experiments to determine the most reliable method of measuring the flow of air and to investigate the flow of air in pipes. Three methods of measurements were tested: 1st, anemometers; 2d, Pitot tubes; 3d, circular and square orifices. The volume of air was accurately measured by being led into a gas-holder of known capacity, and comparisons were then made between the volume measured in the holder and the volume calculated from the measurements made with the instruments.

Two series of experiments were made, one in 1884, in which the pressure on the holder was 2¾ inches of water, and a second series in 1885, having a pressure on the holder of 4½ inches of water. The conclusions reached were that the anemometers gave readings from 7 to 13 per cent in excess, but that anemometers, Pitot tubes and orifices gave readings that were sufficiently accurate if proper corrections were made and proper coefficients were used in determining the quantity of air. It was found that the mean speed of the air was obtained at two-thirds of the radius out from the center of the discharge pipe. The speed and pressure were maximum at the center of the pipe and miminum at the inner circumference.

With regard to the resistance to flow of the air in the pipe it was found that for cast-iron pipes the resistance to flow increases as the square of the speed of the air and as the two-thirds power of the density of the air.

The most satisfactory method for measuring the flow of air is by means of the Pitot tube, which is arranged as follows: Consider a current of air to be flowing through the pipe A, Fig. 88, in the direction of the arrow. A curved tube B, open at the end, is introduced through the side of the pipe with its opening

turned in the direction from which the current flows. Another tube *C*, which is straight and open at the end, is also introduced through the side of the pipe. These tubes *B* and *C* are connected by suitable pipes to a glass U tube, the curved tube *B* being connected to one end of the U and the straight tube *C* to the other end.

If the U tube be partially filled with water, the water will rise in that leg of the U which is connected to the straight tube *C* and descend in the leg connected to the curved tube *B*. The

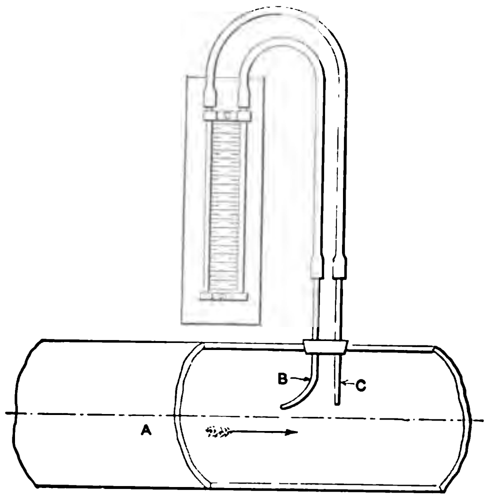


Fig. 88.

difference in level will be the head of water due to the velocity of the flow of air in the pipe *A*. The method of using the head so found is explained in connection with the test made by the De Laval Steam Turbine Co.

An account is given in vol. CXXII of the Proceedings of the Institution of Civil Engineers of a number of experiments on fans by Mr. Bryan Donkin. These experiments were performed for the purpose of finding the pressure, volumetric and mechanical efficiencies of the fans tested. Eleven fans were tested, measurements being made by means of Pitot tubes, and the discharge being through orifices of known dimensions formed by perforating zinc plates. The accompanying table gives the summary of the best experiment on each fan.

SUMMARY OF THE BEST EXPERIMENT ON EACH OF THE ELEVEN FANS TESTED BY MR. BRYAN DONKIN.

Number of Fan.	Type of Fan. C.I. indicates Cast Iron. W.I. indicates Wrought Iron.	Diameter over Vanes, in Inches.	Revolutions of Fan per Minute.	Static Pressure at Outlet of Fans, Inches of Water.	Dynamic Pressure before Baffle, Inches of Water.	Quantity of Air Delivered, in Cubic Feet per Minute.	I.H.P. of Fan only.	Theoretical H.P.	Mechanical Efficiency, Per Cent.	Pressure Efficiency, Per Cent.	Volumetric Efficiency, Per Cent.	Equivalent Orifice, Square Foot.	Conditions of Baffling and Direction of Vanes to the Outlet.	Temperature of Air at End of Pipe, Wet Thermometer and Dry Thermometer, ° F.	Barometric Pressure, Inches of Mercury.
I	Twenty vanes W.I., 1½ inches wide at tip.	19⅜	1291	3⅞	4⅞	2700	3.44	2.04	59.40	87.00	60.38	0.44	One sheet perforated zinc Concave.	44 wet 54 dry	30.2
II	Twelve vanes, C.I., 2½ inches wide at tip.	25½	1354	4½	4⅞	1905	3.31	1.47	44.49	48.68	18.65	0.33	Two sheets zinc. Concave.	75 wet 77½ dry	30.08
III	Six vanes W.I., 7 inches wide at tip.	23¼	1248	4¼	4 7/16	1657	1.93	1.18	60.92	63.20	23.34	0.30	Two sheets zinc. Radial.	74 wet 82½ dry	30.36
IV	Eight vanes W.I., and eight half vanes 4⅛ inches wide at tip.	23⅝	1014	4 3/16	5 3/16	1261	3.16	1.05	33.13	52.65	20.75	0.21	Three sheets zinc Concave.	55 wet 68 dry	29.80
V	Twenty-four vanes W.I., 3½ inches wide at tip.	15¾	1500	3⅓	3 11/16	2286	2.40	1.35	56.35	79.80	85.90	0.45	One sheet zinc. Concave.	88 wet 91 dry	30.28
VI	Six vanes W.I.	20	1589	3⅞	4¼	1636	2.39	1.11	46.59	50.49	28.33	0.30	Two sheets zinc. Convex.	69 wet 74 dry	29.70
VII	Six vanes W.I., 3¼ inches wide at tip.	24 13/16	1355	3 7/16	3¾	1526	2.47	0.91	36.95	39.02	15.93	0.30	Two sheets zinc. Convex.	61 wet 69 dry	30.20
VIII	Six vanes W.I., 1¾ inches wide at tip.	24	1352	....	3 9/16	1163	1.66	0.66	39.57	39.59	13.82	0.24	Two sheets zinc. Convex.	56 wet 61 dry	30.50
IX	Four vanes C.I., 2¾ inches wide at tip.	24⅜	1669	5	5 11/16	1074	2.11	0.97	46.10	38.88	9.48	0.16	Four sheets zinc Concave.	55 wet 63 dry	30.34
X	Eighteen vanes, twelve small and six large, 1 3/16 inches wide at tip.	23⅝	1747	9⅛	8⅞	1280	3.22	1.82	56.57	59.25	12.22	0.17	Three sheets zinc Straight.	57 wet 61 dry	30.00
XI	Ten vanes, C.I., ¾ inch wide at tip.	15¾	2097	1 5/16	1 5/16	913	0.66	0.19	29.34	14.31	25.17	0.31	Two sheets zinc. Concave.	57 wet 63 dry	30.52

Mr. Donkin also states some general conclusions which were based on the results of the experiments. He says that: "sufficient attention is often not given to the admission of air to the center of the fan to reduce friction. The number and shape of the vanes and their direction of rotation seem often to have been guessed at and not deduced from experimental research, which is still needed to decide the latter question for a given speed, quantity and pressure of air. Between twenty and twenty-five vanes give the best results. The shape of the blades, their number and the space between them and the outer casing exercise a considerable influence on the various efficiencies. The final inclination, or angle, of the vanes at their circumference has more effect on the pressure of air and less on the mechanical and volumetric efficiencies. The revolving portion of the fan should always be accurately balanced. . . . The friction of the air inside the casing is often excessive and care should be taken to allow its entrance and passage through the vanes, and out of the fan, with a minimum of skin friction. Changes of direction and shocks which produce losses of head with the high velocities of air should be avoided as much as possible."

Some of the fans tested by Mr. Donkin were run both in the direction which the makers intended them to run and in the reverse direction. In several instances the fans were found to be more efficient when the fan-wheels were reversed or run in the opposite direction to that specified by the makers.

The I.H.P. of the fan was found by indicating the engine and deducting the power required to drive the engine, belt and a fan-shaft that carried no blades. The formulas used for finding the various efficiencies are given by Mr. Donkin as:

Let H = head in meters of air;
h = pressure in millimeters of water;
w = weight of one cubic meter of air in kilograms at the atmospheric pressure and temperature;
V = velocity of air, in meters per second, at end of discharge pipe;
Q = quantity of air delivered, in cubic meters per second;
g = acceleration due to gravity;
U = speed of tips of vanes, in meters per second;
r = radius of vanes, in meters;

$$H=\frac{h}{w},\qquad V=4\sqrt{h}.$$

$$\text{Theoretical H.P.}=\frac{Q\times w\times H}{75}.$$

$$\text{Mechanical efficiency}=\frac{\text{Theoretical H.P.}}{\text{I.H.P. for fan only}}.$$

$$\text{Pressure efficiency}=\frac{g\times H}{U^2}.$$

$$\text{Volumetric efficiency}=\frac{Q}{U\times r^2}.$$

$$\text{Equivalent orifice (in square meters)}=\frac{Q}{0.65\sqrt{2gh\times\frac{1000}{w}}}.$$

The coefficient 0.65 in the formula for the equivalent orifice has been found by many experiments to be the coefficient of discharge for a circular orifice in a thin plate.

Although Mr. Donkin gives the formulas in metric units, the results given in the table are given in English units. The formulas can be used for measurements made in English units by taking the units that correspond to those given in the formulas and changing the constants to keep the proper relations.

High-pressure Fans.—Fans for very high pressures have recently been constructed both in America and Europe. One of the foremost designers of such fans is Prof. Rateau, of St. Etienne, France, who has made many experiments on the driving of fans by turbines. In one set of these experiments he ran a fan having a diameter of 10 inches at speeds varying from 8000 R.P.M. to 20,200 R.P.M. The fan was driven directly by a steam-turbine of the Pelton-wheel type 11.8 inches diameter. The efficiencies obtained for the turbine and fan together varied from 12% to 29%. It was found by separate experiments, that the efficiency of the turbine was about 50%, so that the fan alone, under the best conditions, at about 1650 R.P.M., gave an efficiency of nearly 60%.

Some very interesting tests of high speed fans have been made in this country at the works of the De Laval Steam Turbine Co., for the results of which the author is indebted to Mr. Albert E. Guy. These tests were made on a No. 7 Sturtevant blower connected to a De Laval 20-H.P. steam-turbine running

at 2000 R.P.M. The blower and turbine were mounted on the same bed-plate, as shown by Fig. 89. The wheel of the blower is shown at the left of Fig. 90.

The purpose of the test was to determine: 1st. The pressure produced by the blower. 2d. The amount of air discharged, and the air horse-power (A.H.P.). 3d. The characteristic of the blower by means of varying the pressure or the quantity

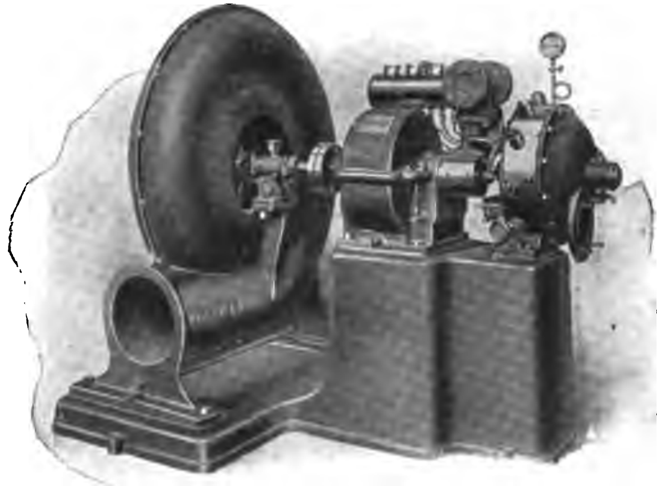


FIG. 89.

discharged. 4th. The steam consumption per A.H.P., and the efficiency of the blower.

In order to obtain these results as accurately as possible, a wooden box 48″×42″×52″ was constructed and connected to the discharge opening of the blower. On the top of the box a circular opening was made, over which the different size converging air-nozzles were placed. In front of the nozzles a Pitot tube of small diameter, connected to a water-gauge, was attached in such a way that its receiving end, flush with the discharge edge of the nozzle, could be moved across the whole diameter of the nozzle. This was done in all the various tests; the variations in the Pitot gauge reading at various points of the nozzle were noted and an average value determined. The readings on the Pitot tube gauge registered the head due to the discharge velocity in the nozzle.

The pressure produced by the blower was ascertanied by means of a water-gauge connected to the box in a place where it was reasonable to suppose that the readings would not be influenced by the velocity of the air.

Six separate tests were made, five with the blower discharging through nozzles 4, 5, 6, 7, and 9 inches in diameter, and the sixth test with the nozzle entirely closed so that there was no discharge.

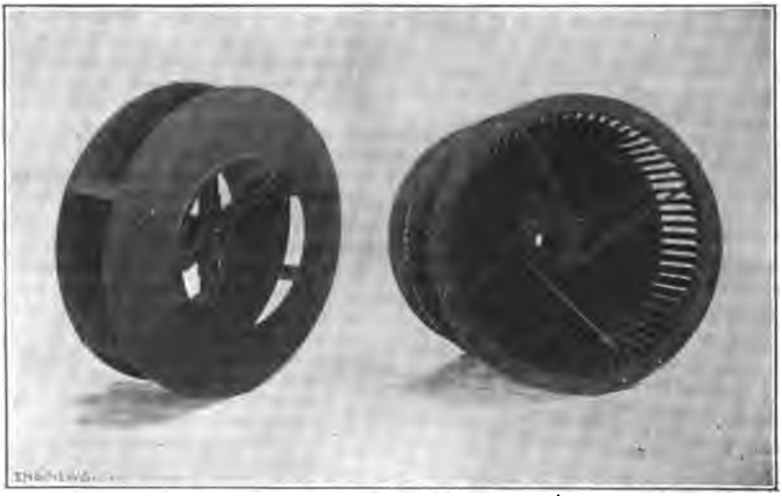


FIG. 90.

As the object of the tests was primarily to obtain information in regard to the blower, the steam consumption of the turbine, per brake horse-power, is much higher than would be the case were it run under more favorable conditions.

Besides the readings of the pressure and Pitot tube gauges, the following observations were taken during each test: The pressure of the atmosphere; the temperature, relative humidity of the air and vapor pressure, by means of a wet and dry bulb thermometer; the temperature in the air box; the steam-pressure above and below the governor-valve; the temperature of the steam; the number of revolutions per minute, and the steam consumption. This last was determined by means of a nozzle test, wherein a nozzle of the same shape and size as those used in the turbine was connected to the turbine-wheel

case by means of a pipe through which steam was discharged into a barrel of water placed on a scale. Between this nozzle and the barrel a steam-gauge and a three-way cock were placed, the former to ascertain that the pressure on the discharge side of the nozzle was at all times below 58% of the pressure in the wheel-case. This pressure at no time rose above 43 lbs., thus satisfying the requirements of an accurate test. The three-way cock was set, immediately before the test, to discharge the steam into the atmosphere so as to remove all condensation from the pipe, then quickly turned so as to discharge the steam into the barrel. The time during which the steam was discharged into the barrel was carefully noted and the weight of water in the barrel was taken before and after the discharge. The difference in these weights then gave the total steam-consumption per nozzle for the time of the nozzle test, which was three minutes for each test.

The brake horse-power of the turbine was determined in the following manner: After the blower had been tested it was removed and a prony brake substituted, which was loaded

TEST OF DE LAVAL STEAM TURBINE

September 26

Steam Pressure at the Gov.-Valve, Lbs. per Square Inch.		Temp. of Steam, F.°	Total Steam used by Turbine, Lbs.	Revolutions per Min.	Steam Nozzle Open No.	Thermometer Readings, F.°		Temp. of Air in Box, F.°	Baro. Press. Inches Mercury.	Diam. of Air Nozzle Inches
Above.	Below.		Hour.			Wet.	Dry.			
										Test
101.75	58.75	335.5	752	2015	2 , 5	72.9	86.5	98.6	29.91	4
										Test
101.75	64.5	351	810	2023	2 , 5	76	88	100	29.91	5
										Test
101.75	68.75	343	854	2033	2 , 5	72.9	87.4	100	29.9	6
										Test
101.75	74	348	906	2042	2 , 5	72.65	87.15	100	29.91	7
										Test
101.75	90	335.4	1065	1990	2 , 5	74.2	83.5	93.8	30.02	9
										Test
....	51.2	335.5	....	2053	2 , 5	......	......	93.8	....	......
1	2	3	4	6	7	8	9	10	11	12

Steam-nozzles used: No. 2, Diam. —

until the steam-gauge below the governor-valve registered the same pressure as during each of the blower tests. The speed and number of nozzles open were also adjusted to the conditions existing during each of the blower tests. This method of obtaining the brake horse-power and the efficiency of the blower was also used for testing a number of De Laval centrifugal pumps and is very reliable.

The accompanying table shows the final results, which are further illustrated in the characteristic. The turbine was operated non-condensing.

In order to obtain accurate results, for comparison, when testing blowers, it is particularly necessary to make observations of the atmospheric pressure and temperature of the air and also of the moisture it contains, although, under ordinary conditions of the atmosphere, the latter will not affect the results very materially.

The weight of one cubic foot of air is dependent upon the atmospheric pressure, temperature and moisture and is

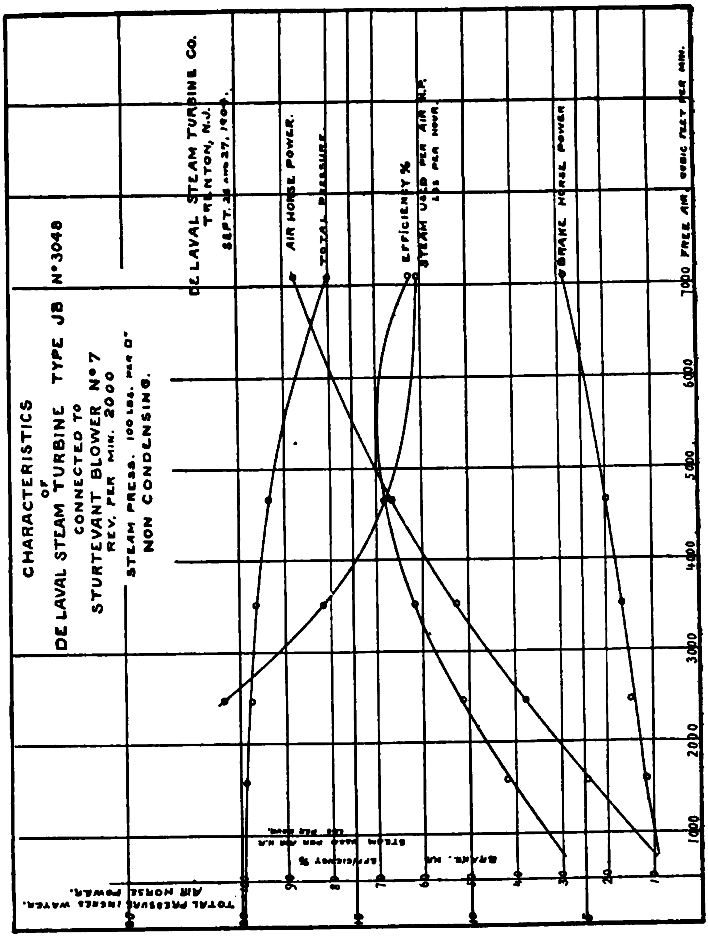
$$W=\frac{0.0807(p-0.379he)}{(1+\alpha t)29.921},$$

STURTEVANT BLOWER NO. 3048.

and 27, 1904.

Pressure, Inches of Water in		Quantity of Air, Cu. Ft. per Min.		Air Horse-power.		Brake Horse-power.	Efficiency of Blower, Per Cent.		Steam used per A.H.P. Hour.	
Air Box.	Pitot Gauge.	For-mula.	Pitot Gauge.	For-mula.	Pitot Gauge.		For-mula.	Pitot Gauge.	For-mula.	Pitot Gauge.
No. I. 19.78	19.85	1584.7	1585	4.87	4.88	11.625	41.89	41.97	154	153.8
No. II. 19.47	19.438	2475.9	2456.2	7.63	7.51	14.625	51.17	51.35	106	107.8
No. III. 19.01	18.845	3511.7	3512.9	10.42	10.43	16.875	61.7	61.8	81.9	81.7
No. IV. 18.71	18.175	4656.9	4659	13.3	13.38	19.625	67.8	68.2	68.1	67.7
No. V. 15.988	15.625	7051.7	7100	17.1	17.5	28.06	61.2	62.3	62.2	60.8
No. VI. 19.875	No	discha	rge.							
13	14	15	16	17	18	19	20	21	22	23

0.327 inch; No. 5, Diam. = 0.382 inch.



where 0.0807 = the weight of one cubic foot of dry air at 32° F. and at a normal atmospheric pressure of 29.921 inches;

0.379 = the percentage of vapor contained in one pound of saturated air at 32° F.;

p = atmospheric pressure, in inches of mercury;

h = relative humidity;

e = vapor-pressure, in inches of mercury;

α = coefficient of expansion for air;

$(1+\alpha t)$ = the volumetric ratio of dry air at 32° and t° temperature.

Thus: for test IV we find from the table of relative humidities for readings given by the dry- and wet-bulb thermometers,

$$h=0.49, \quad e=1.287, \quad t=55.15°, \text{ and } p=29.91,$$

from which

$$W=\frac{0.0807[29.91-(0.379\times0.49\times1.287)]}{1.11245\times29.921}=0.0721 \text{ lbs.}$$

The velocity of discharge is $V=\sqrt{2gH}$ ft. per second, H being the head in feet of air, which is derived from the Pitot gauge reading and W, and is equal to the weight of one cubic foot of water divided by the weight of air and multiplied by the Pitot gauge reading, in feet. Thus:

$$H=\frac{62.355}{0.0721}\times1.514=1309.3 \text{ ft.}$$

and

$$V=\sqrt{2\times32.18\times1309.3}=290.29.$$

The amount per air discharged of second is

$$Q=cAV,$$

where A = area of nozzle, in square feet;
c = a constant for the nozzle.

This constant is given by Weisbach and Grashof as 0.097 for a converging nozzle ⅜ inch in diameter. As this constant undoubtedly varies with the diameter of the nozzle, approaching unity with increasing diameter, and as it is always difficult to determine, the test is simplified if its value be taken as unity, as has been done here.

Then for test IV we have:

$$Q=.2672\times290.29=77.65 \text{ cu. ft. per second,}$$

or 4659 cu. ft. per minute.

The work L expended upon the air is equal to the work done by the expanding air. Hence, $L=\frac{V^2}{2g}$ ft.-lbs. and the air horse-power, when discharging Q cubic feet per second, is

$$\text{A.H.P.}=\frac{0.0721\times77.65\times(290.29)^2}{2g\times550}=13.38.$$

During test IV the brake horse-power was found to be 19.625; hence the efficiency of the blower is:

$$E=\frac{13.38}{19.625}=68.2\%.$$

The total steam-consumption was 906 lbs. per hour and the steam-consumption per air horse-power 67.7 lbs. per hour.

The following formula for the velocity of discharge from a nozzle affords a check on the Pitot tube method.

$$V=2g\frac{n}{n-1}p_0v_0\left[\left(\frac{p}{p_0}\right)^{\frac{n-1}{n}}-1\right],$$

$g=32.18$ ft. per second;
p_0 = pressure of air outside box, in pounds per square foot;
v_0 = specific volume of air at pressure p_0;
p = pressure of air in box, in pounds per square foot;
$n=1.41$.

The following results show a comparison of the two methods for determining the velocity V:

Number of Test.	V, by Formula.	V, from Pitot Gauge.
I	302.55 ft.	302.60 ft.
II	302.20 ft.	301.00 ft.
III	297.10 ft.	297.20 ft.
IV	290.15 ft.	290.29 ft.
V	265.90 ft.	268.10 ft.

This sufficiently demonstrates the reliability of either of the two methods for determining V.

The accompanying curve shows the results obtained from the Pitot tube readings, but in the table, columns 15, 17, 20 and 22, are given the results calculated from the formula for V.

These blower sets are arranged with various fans, Fig. 91, showing a 150-H.P. De Laval Steam Turbine directly connected

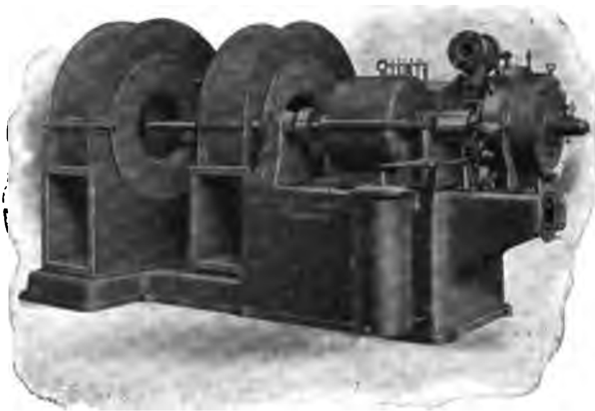


FIG. 91.

to a Sirocco blower, the double intake wheel of which is shown at the right in Fig. 90. They are principally used in the manufacture of gas, for obtaining the proper temperature in the gas generator. They are so compact that they can be readily adapted to many purposes, particularly in marine work where their perfect balance, insuring absence from vibration, makes them an ideal set for blowing and ventilating purposes.

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